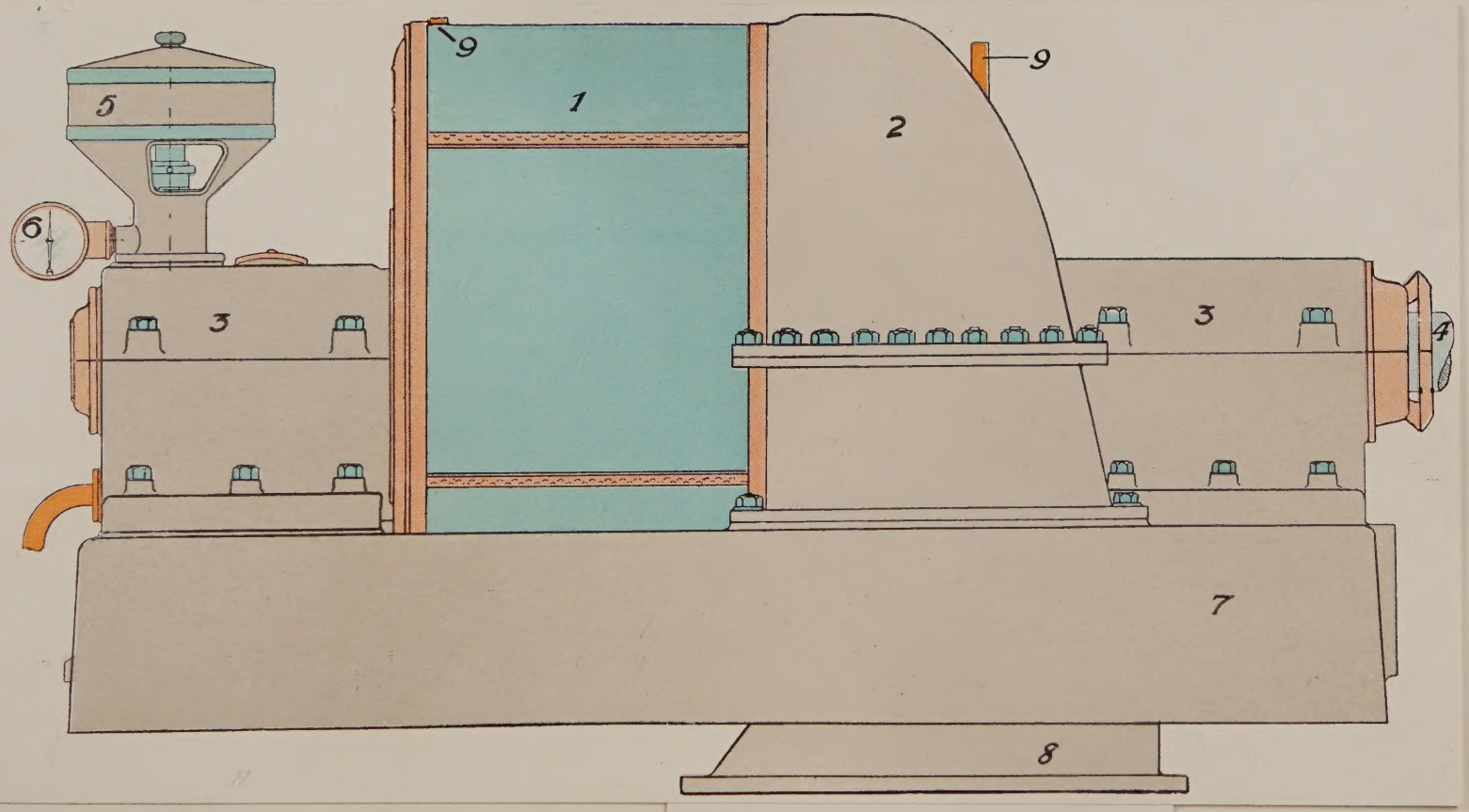
MODERN POWER ENGINEERING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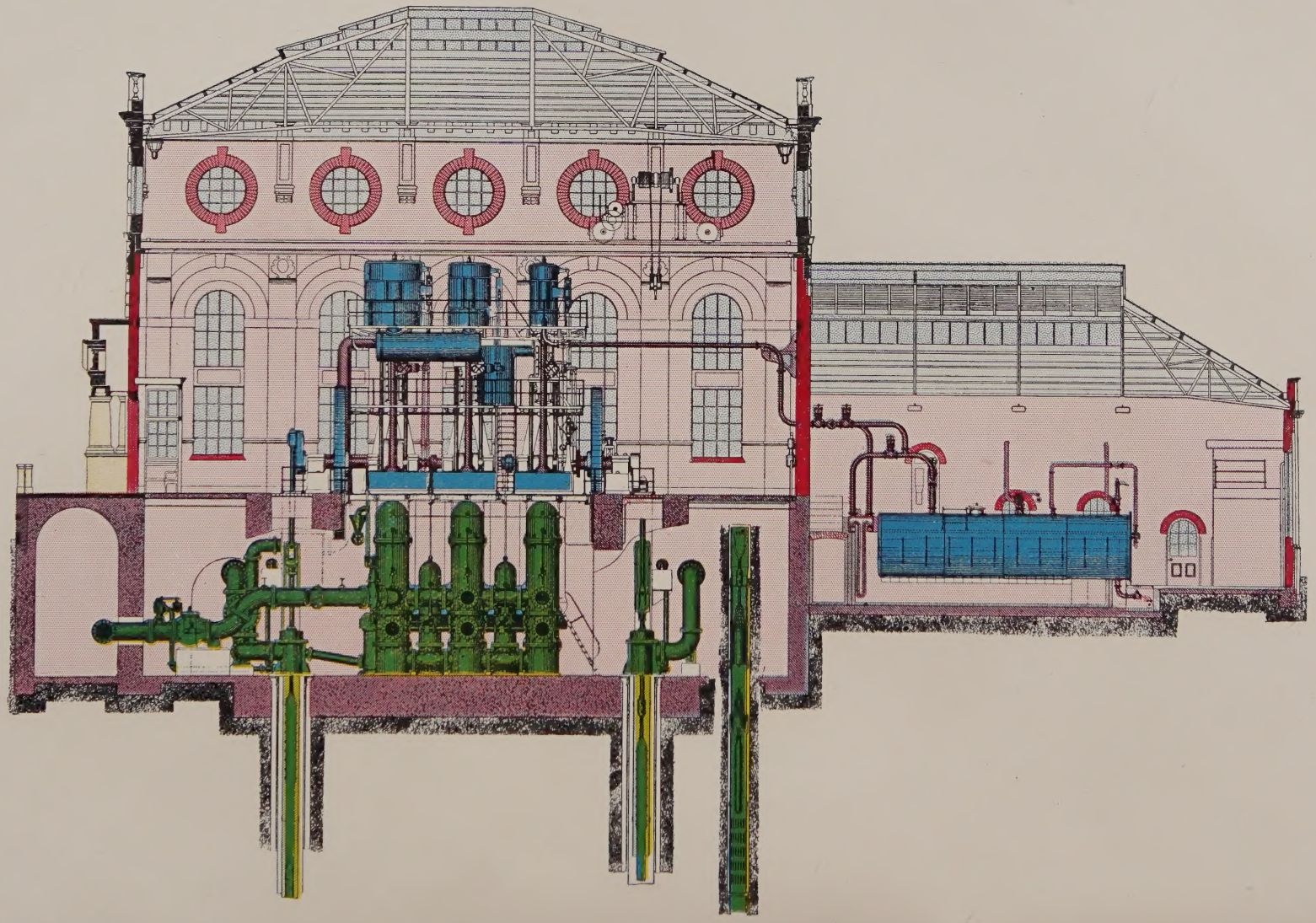
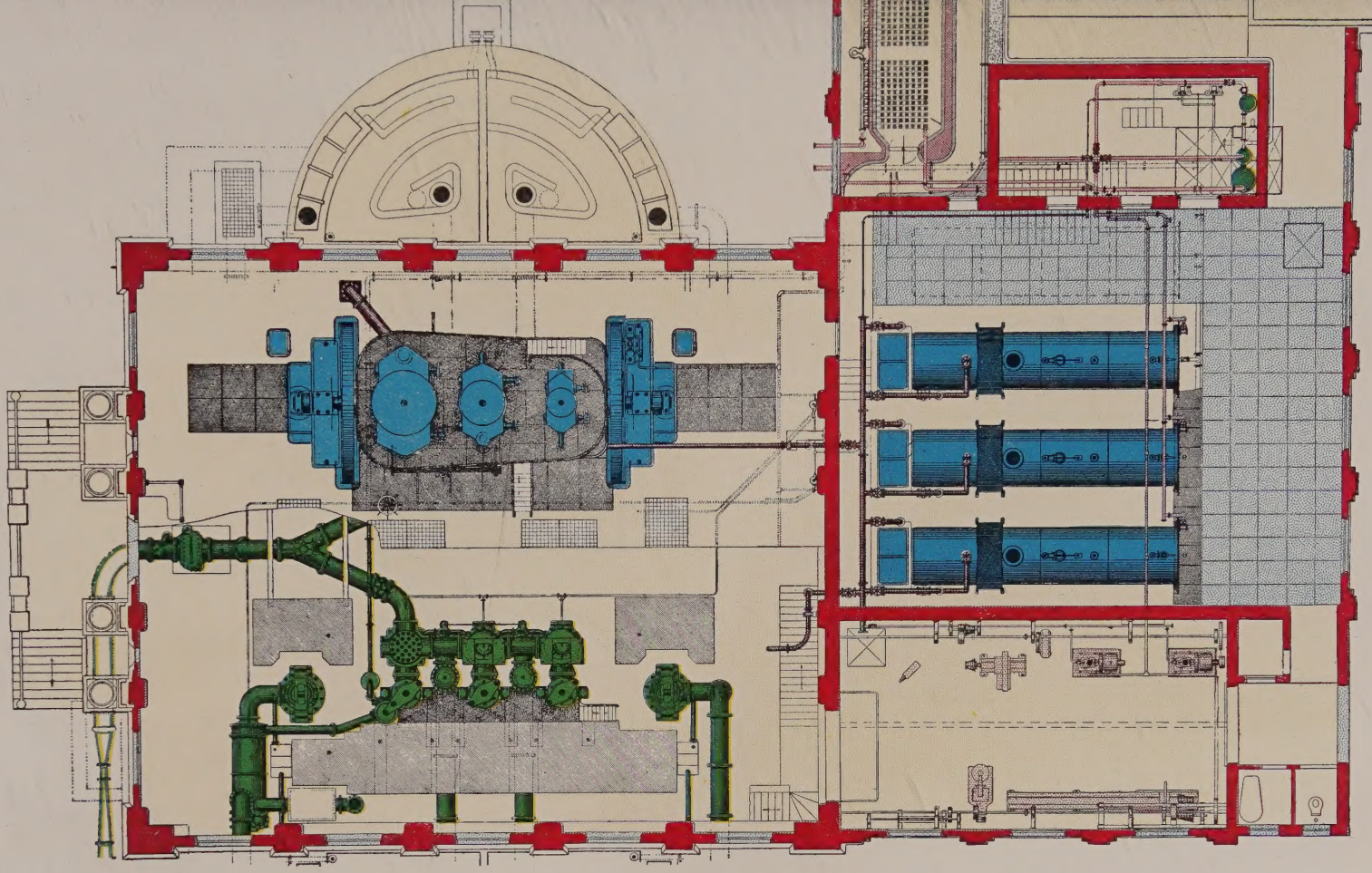
KEY TO MODEL OF STEAM TURBINE

1. Turbine Casing.
2. Exhaust Casing.
3. Main Bearing Housings.
4. Shaft.
5. Governor.
6. Tacometer.
7. Bed Plate.
8. Exhaust Outlet.
9. Steam Sealing Pipes.
10. High Pressure Nozzle Casing.
11. Shaft Glands.
12. Insulating Material.
13. Thrust Block.
14. Main Bearing.
15. Gland Packing.
16. High Pressure Nozzle.
17. Velocity Wheel.
18. Diaphragms.
19. Blade Discs.
20. Nozzles.
21. Blades.
22. Flexible Shaft Coupling.
23. Governor Driving Gear.
24. Planished Steel Lagging.

MODERN POWER ENGINEERING





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BY

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WITH MANY HUNDRED ILLUSTRATIONS

VOLUME III

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PREFACE

DESPITE the severe competition which it has to face from the steam turbine on the one hand and from the various forms of internal combustion engine on the other, the reciprocating steam engine is still a very important form of prime mover and one which is used very extensively in the commercial and the industrial world to-day. Having dealt in the earlier volumes of this work with those two newer forms of engine mentioned above which have so seriously challenged the supremacy of the old reciprocating steam engine, the author has made the latter type of prime mover the principal subject of this volume. In addition to stating the principles underlying the design and operation of the "reciprocator" as it is sometimes called, descriptions of the various forms which are being built at the present day have been included. It is indicative of the importance of the reciprocating steam engine that a very large number of firms are still engaged in the production of this type, and among these the author is indebted to the following firms who have so generously placed particulars, drawings, and photographs of their engines at his disposal: Messrs. Robey & Co., Ltd., Messrs. Peter Brotherhood, Ltd., Messrs. Belliss & Morcom, Ltd., Messrs. Galloways, Ltd., Messrs. Glenfield & Kennedy, Ltd., Messrs. Worthington & Simpson, Ltd., Messrs. Ashton, Frost & Co., Ltd., The Hydraulic Engineering Co., Ltd., Messrs. Richardsons, Westgarth & Co., Ltd., The Wallsend Slipway & Engineering Co., Ltd., The North British Locomotive Co., Ltd., The Sentinel Wagon Works (1920), Ltd., and the Baldwin Locomotive Works, Ltd., of Philadelphia, U.S.A. Acknowledgments are also due to Marine and Locomotive Superheaters, Ltd., and also to the editors of *Shipbuilding and Shipping Record*, *The Marine Engineer*, and *The Railway Gazette*, for certain particulars and trial results which have been embodied in the chapters on marine engines and locomotives. In the chapter on pumps attention has not been confined exclusively to the reciprocating pump, and the authors acknowledgments are due to Messrs. Mather & Platt, Ltd., for much valuable information regarding their rotary pumps, and also to Messrs. William Beardmore & Co., Ltd., for particulars of the Humphrey Gas Pump.

Space has also been found in this volume for a consideration of the vastly important subject of water power, and in addition to describing the principles underlying the development of water power descriptions of three typical hydro-electric installations in this country have been included, in addition to particulars of a number of notable power plants abroad. The author is indebted to S. E. Britton, Esq., M.I.E.E., the City Electrical Engineer, for particulars of the Chester installation, to the English Electric Co., Ltd., for particulars of the installation at York, and to Messrs. Escher Wyss et Cie of Zurich, not only for the particulars of the Kinlochleven installation, but also for details and illustrations of a number of other installations abroad. His acknowledgments are also due to Messrs. Boving & Co., for particulars of the Tasmania plant and for details of the Kaplan turbine. Finally he must express his indebtedness to the Hydro-Electric Power Commission of Ontario, Canada, who, in addition to sending the official particulars of the various installations on the Niagara Falls, supplied the various drawings and photographs of the plant.

A. R.

February 1924

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MODERN POWER ENGINEERING

VOLUME III

CHAPTER I

THE RECIPROCATING STEAM ENGINE—A BRIEF EXPLANATION OF HOW IT WORKS

IF the reader will glance back to the history of the steam engine which we gave in an early chapter in Volume I of this work, he will find that although one of the first forms of steam engine of which we have any authentic record was of the rotary type, it was with the discovery of the piston type of engine that the first real step in the direction of the practical application of the power of steam to the service of man was made. In the steam engine as developed by Watt, the pressure of the steam is employed to give motion to a piston which moves in a straight line along the axis of a cylinder, and by some means or another this straight line motion is converted into a motion of rotation at the flywheel of the engine. In the earlier forms, the steam acted on one side only of the piston; but in the latter types the steam acted alternately first on one side of the piston and then on the other, thus imparting an upward and a downward motion, or in other words a "reciprocating" motion, to the piston. It will be recalled that the first piston engines had vertical cylinders, the reciprocating motion of the piston being converted into the motion of rotation of the shaft by means of a beam supported in bearings at its middle point. Later, the beam was dispensed with, and while vertical cylinders were often used, other positions of the cylinder, either horizontal or inclined, were adopted, but the reciprocating motion of the piston in the cylinder was, of course, still retained. The piston engine in its many forms was, until the rediscovery of the steam turbine by Sir Charles Parsons and Gustav de Laval, the only form of steam engine that was available for commercial purposes, and hence it was called briefly the steam engine without any qualification whatever. With the development of the steam turbine, however, it became necessary to distinguish between the two forms of engine that were then on the market, and thus the piston engine gradually became known as "the reciprocating engine." The turbine, it may be noted, is often referred to as the rotary engine, since all its moving parts have a motion of rotation, thus distinguishing it from the reciprocating engine.

The reciprocating steam engine, while it has been to a very large extent displaced by the steam turbine, is still from the commercial point of view a very important form of power generator. It might almost be urged that the development of the steam turbine is due not so much to its inherent superiority to the reciprocating steam engine as to the enormous growth in the use of electricity for industrial purposes. Whether the progress in steam turbine design has been responsible for the remarkable developments that have occurred in the use of electrical energy for lighting and power purposes or whether the reverse is the case—that is, whether progress in electrical engineering has fostered the development of the steam turbine—is a subject which is open to question;

but the fact remains that the past thirty years have witnessed the development of both the steam turbine and the electrical industry. For the generation of electricity high speeds are desirable, and as the steam turbine is essentially a high-speed engine, it found its most ready application to the driving of electrical generators. But where slower speeds were required, as, for example, in the driving of steel rolling mills or for driving the pumps which supply the water to our large towns, there the reciprocating engine is still widely adopted. Again, the steam turbine, while it is sometimes employed for the generation of small powers, is essentially an engine for the development of large powers. At low power it is very inefficient, and, moreover, the speed of rotation of small turbines is even higher than that of large turbines. Hence for the development of small powers, particularly when a low speed of rotation is desirable, the reciprocating steam engine is found to be more suitable than its younger rival. Hence we find that for many purposes, such as driving steel rolling mills, operating the pit-head winding gear at the mouth of coal mines, driving small electrical generators, etc., the reciprocating engine is still largely used.

There is yet another field in which the reciprocating steam engines find almost universal application, and that is in the driving of railway locomotives. We have already seen that attempts have been made to employ the steam turbine for driving locomotives, but in addition to the disadvantage of high speed, the turbine is further handicapped by the fact that it cannot be reversed. In railway work it is very necessary that the locomotive should be capable of being run equally well in either direction—that is, the engine driving it must be capable of being easily reversed, and from this point of view the reciprocating steam engine is second to none among all the known forms of power generators. The electric motor can be very readily made to run in either direction, but means have to be found for supplying electrical energy to the motor, and hence from the point of view of railway working the electric locomotive can only be used under circumstances which warrant the laying down of transmission lines for the entire length of the track from power stations situated at convenient points. The self-contained locomotive carrying its own source of energy in the form of either coal or oil will for very many years to come yield the most suitable means of operating long-distance railway lines passing through thinly populated stretches of country. Perhaps it may be found advantageous to employ a steam turbine to drive an electric generator, this generator supplying current to operate electric motors, these in their turn driving the locomotive ; or it may even be that some form of Diesel engine will be installed, thus doing away with the steam boiler, the Diesel engine either driving the locomotive direct or else driving an electric generator to supply current for the electric driving motors. One might even go so far as to suggest the possibility of an oil turbine driving a locomotive either directly or by means of electric motors. But we have no desire to prophesy. The fact remains that, despite the development of the steam turbine and the internal combustion engine, the reciprocating steam engine is almost universally employed for driving locomotives on railways, except where electric traction is adopted. The steam locomotive is probably the most inefficient form of prime mover which is used on a commercial basis at the present day ; but so great are the advantages of the steam reciprocating engine, its moderate speed of rotation, its flexibility, and its power of being easily reversed, that it still stands pre-eminent for this particular service.

The problem of driving ships may at first sight appear to be very similar to that of driving locomotives, and to a certain extent such is the case. It is essential, except in the fastest vessels, that the speed of the propeller should be very slow, and therefore if the engines are directly coupled to the propeller shaft, slow-speed engines must be employed. Indeed, the speed of rotation of the propeller of a ship is in the majority of cases considerably less than is the speed of rotation of the driving wheels of a locomotive,

the former being below 100 revs. per min. in the case of cargo vessels as against about 275 revs. per min. for a passenger locomotive. But the length of the voyage of a vessel may be reckoned in weeks where the journey of a train is reckoned in hours, and apart from questions of speed of rotation, a marine engine has to be constructed so as to be capable of running continuously for many days. The author has himself been an engineer on board a ship in which the engines each voyage used to run for twenty days without a stop, and this is by no means an exceptional performance. It signifies two things, however—first, that very great reliability is essential in a marine engine, and second, that the capability of being able to run equally well in either direction, with which may be coupled the necessity for extreme facility in reversing the engines, is almost as essential in a marine engine as it is in a locomotive. There are, however, certain factors which serve to differentiate the marine engine from the locomotive engine, the most important of which is perhaps the considerably greater power which is required in the former case. This factor together with those mentioned above have made it possible to employ the steam turbine for the propulsion of ships, the difficulty of the high speed being overcome by the use of speed reduction gearing, which enables an economical high-speed turbine to be employed in conjunction with an efficient low-speed propeller, while the need for being able to move in either direction is met by the installation of special "astern" turbines, which are only employed when the vessel is manœuvring in and out of harbour or at times of emergency. The power developed by the astern turbines is only a fraction of that developed by the ahead turbines, since they are not doing any useful work during the major portion of the voyage when the vessel is running full speed ahead.

The peculiar requirements of marine service have also been met in a more or less satisfactory manner by the design of various types of heavy oil engine, in which the speed of the engine is generally well suited to the direct driving of the propeller, although it is felt that it would be advantageous if a slower speed of rotation could be economically obtained. The marine oil engine is not, however, without its disadvantages, one of the most serious of which lies in the difficulty which has so far been experienced in obtaining prompt and reliable means for starting, stopping, and reversing. In manœuvring a vessel in or out of harbour the main engines may be called upon to start quickly in either direction and as quickly to stop again; moreover, there may be long periods during which the engines must run dead slow. From the point of view of ease of control the marine Diesel engine leaves much to be desired, although developments are continually being made in the direction of improving the effectiveness of the control gear. There are other disadvantages inseparable from the use of this type of engine which tend to militate against its extensive adoption for the propulsion of ships to which attention might be drawn, but these will be well understood if the reader has carefully studied the chapters in Volume II dealing with the internal combustion engine, in which its limitations have been set forth in detail.

Thus, despite severe competition which it has to meet from its younger rivals, the steam turbine and the oil engine, the reciprocating steam engine is still employed on a very large scale for marine purposes. Where large powers are required, as for ocean liners or war vessels propelled at high speeds, there the steam turbine is almost exclusively adopted; but for vessels of moderate power, the cargo vessels and the tramp ships, which form the bulk of our vast mercantile marine both as regards the number of such vessels and their aggregate tonnage, there we find that the reciprocating steam engine is even now almost always installed. To recall the figures from *Lloyd's Register of Shipping*, to which reference has already been made, 77 per cent. of the steamers classified in *Lloyd's Register* are propelled by reciprocating steam engines, and it is safe to say that by far the larger proportion of the remaining 23 per cent. represent the larger passenger

vessels, cross-Channel steamers, etc., which are propelled by steam turbines. Mention may also be made of the fact that on paddle steamers—and there are still a large number of vessels of this type engaged in the summer excursion traffic and for passenger work generally in sheltered waters—the reciprocating steam engine is invariably used for driving the paddle wheels.

Sufficient has, however, been said to prove that, despite the development of newer and in many respects better forms of power generators, the reciprocating steam engine is still of very great importance in the industrial world, and in its different forms it is used very extensively at the present day. Having in the previous volumes of this work described numerous forms of steam turbines and of internal combustion engines, we must now devote our attention to the older but still very important reciprocating steam engine, remembering that this type is still being extensively built, for the various purposes which we have briefly set out above, and that some of the more notable examples of engines of this type can justly claim to be included in our work on "*Modern* Power Engineering." Before doing so, however, we must first of all give a brief outline of the action of the reciprocating steam engine in order that the peculiar features of the different types we describe can be judged at their true value. A simple diagrammatic arrangement of a reciprocating steam engine is shown in Fig. 1. The cylinder C is provided at each end with "ports" or openings, SP and EP, through which the steam can pass to or from the cylinder. The passage of the steam through the ports is controlled by a valve or valves which are not shown on the diagram, although the method of controlling the steam supply will be discussed later. The cylinder is fitted with a piston, P, which is made an easy sliding fit in the cylinder, that part of the piston which comes into contact with the cylinder walls being provided with some form of "packing" which, while minimising the friction, nevertheless prevents the leakage of steam from one side of the piston to the other. The piston is carried by a rod, PR, which transmits the motion of the piston to the crosshead, CH, the crosshead being fitted with shoes working in guides which are accurately machined in alignment with the cylinder, so that the crosshead is compelled to move accurately in a straight line, this line being coincident with the centre line of the cylinder. Where the piston rod passes through the cylinder cover, the latter is provided with an arrangement, SB, termed a "stuffing box," which is bored out so as to be concentric with the cylinder. Like the piston, it is fitted with some form of packing in order to reduce the friction to a minimum and at the same time prevent the leakage of steam from the cylinder into the atmosphere. It will be realised that the accurate alignment of the crosshead guides relieves the stuffing box of any force due to the weight of the moving parts or to the thrust caused by the inclination of the connecting rod, CR, and thus assists in the maintenance of a perfect steam-tightness.

The reciprocating motion of the crosshead is converted into a motion of rotation of the crank shaft, CS, by means of the connecting rod which carries a bearing at each end, that at the crosshead being usually referred to as the small end and that at the crank as the big end. (In a vertical engine these are referred to as the top and bottom end bearings respectively.) The crosshead carries what is termed a gudgeon pin, GP, secured rigidly to it, and it is on this pin that the small end bearing works. The crank shaft, as its name implies, is provided with a crank, which means that it carries a pin, CP—which is of the same diameter as the crank shaft itself, and connected to it by means of either one or two crank webs, CW. When two webs are employed, there is generally a crank shaft bearing on either side of the centre line of the engine; but when only one web is employed, obviously it is impossible to fit a bearing on either side of the centre line of the engine, and we have what is termed an overhung crank. In both cases, however, the crank pin is compelled to describe a circle about the crank

shaft as centre, the diameter of this circle being equal to twice the distance between the centres of the crank pin and the crank web. The big end bearing of the connecting rod works on the crank pin, and thus as the piston moves backward and forward in the cylinder imparting an exactly similar motion to the crosshead and to the small end of the connecting rod, a thrust and a pull is alternately transmitted through the connecting rod to the crank pin, thereby causing the crank shaft to rotate. The centre of the crank pin circle is on the centre line of the engine, and as the piston describes its forward stroke, the crank pin moves through the top half of its circle, while the return stroke of the piston causes the crank pin to describe the bottom half of its circle. Hence a complete forward and backward motion of the piston causes one complete revolution of the crank, and the stroke of the piston is equal to the diameter of the crank pin circle, i.e. is equal to twice the length of the crank.

We must now examine in rather closer detail the means whereby a reciprocating motion is imparted to the piston. The length of the cylinder is made such that when the piston is at the ends of its stroke there is a small space between the surface of the piston and the cylinder cover. This space at each end of the cylinder is called the "clearance," and the steam ports, SP, are so arranged that they communicate with the clearance, although the piston can never entirely close the ports. Imagine now that the piston is in the position shown in Fig. 1. Steam at approximately boiler pressure enters

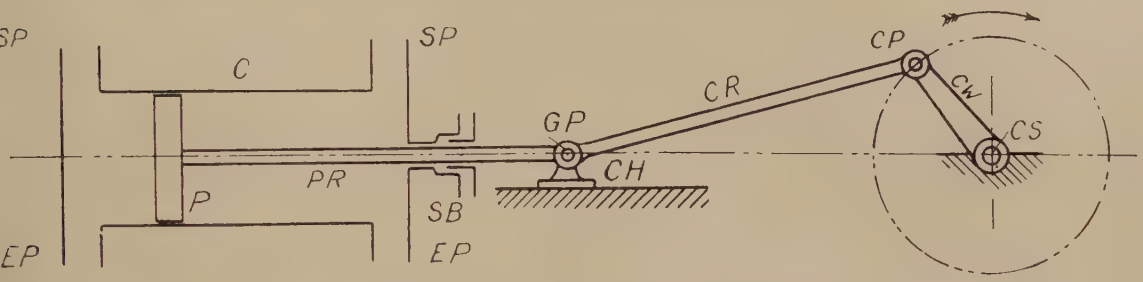


Fig. 1.—Diagrammatic Arrangement of Reciprocating Steam Engine.

the left-hand port, SP, and, pressing on the piston, drives it forward. It must not be imagined that steam is admitted for the entire stroke of the piston, as, while this would undoubtedly cause the piston to move forward, it would prove a most uneconomical method of using the steam. Actually, when the piston has moved through a certain fraction of its stroke the supply of steam to the cylinder is cut off by the movement of a valve which closes the port. The position at which "cut-off" occurs depends upon various circumstances, which will be noted as we proceed; but we may emphasise the fact now, that the earlier the cut-off, the more economically is the steam being employed. After cut-off has occurred, the steam which is imprisoned in the cylinder gradually expands on account of its elastic nature, thus continuing to drive the piston forward for the remainder of its stroke. Just before the stroke is completed, a valve opens the right-hand port, SP, thereby permitting the steam from the boiler to enter on the other side of the piston, thus driving it backward. On this stroke again cut-off occurs at a predetermined position, and for the remainder of the stroke the piston is driven backward by the expansive effort of the steam. Meanwhile when the piston was at the end of its forward stroke, a valve in the lower left-hand port, EP, has opened, thus permitting the steam to pass out of the cylinder. This steam, now called the "exhaust," passes either to the atmosphere, as in the case of the locomotive, or else to a condenser, as in the majority of stationary engines and marine engines, where it is converted into water and returned to the boiler to be reconverted into steam and used over again. When

the piston has reached the end of its backward stroke, the left-hand exhaust port is closed and the corresponding steam port is opened, and at approximately the same time the right-hand exhaust port is opened, thus permitting the steam on the right-hand side of the piston to pass to the exhaust. The alternate admission of steam, expansion, and exhaust on both sides of the piston is continued throughout the entire period that the engine is working.

It will be realised, however, that this alternate backward and forward movement of the piston would impart a very jerky motion to the crank shaft of the engine unless steps were taken to steady the motion and to ensure a uniform and constant speed of rotation. Incidentally it may be remarked that when the piston is at either end of the cylinder, no matter how great the force on the piston may be, it has absolutely no tendency to turn the crank shaft. When the piston is in these positions, the crank and the connecting rod are in the same straight line and the engine is said to be in its "dead centre" positions. In a single-cylinder engine a heavy flywheel must be provided in order to carry the shaft beyond these dead centre positions. On starting up, the engine is moved round, either by hand or by a specially fitted barring gear, to such a position that on admitting steam to the cylinder it thrusts the piston forward with sufficient velocity to enable the flywheel to acquire enough momentum to carry the engine beyond the first dead centre, so that steam entering on the other side of the

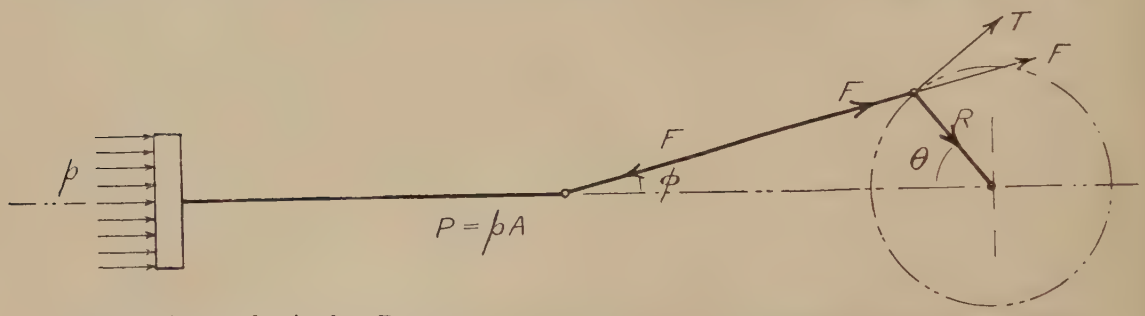


FIG. 2.—Diagram showing how Pressure on the Piston produces a Turning Effort on the Crank Shaft.

piston can thrust it in the opposite direction. The speed gradually rises, some of the energy stored in the flywheel during each stroke being expended in carrying the engine over each dead centre until the normal running speed is attained. The flywheel, in addition to enabling the engine to pass the dead-centre position, also serves the useful purpose of keeping the speed of the engine constant during each successive revolution, despite the variation in the turning effort on the crank. It will be readily understood that while the force on the piston is transmitted along the piston rod, owing to the fact that the connecting rod is inclined at a varying angle to the centre line of the engine, i.e. to the piston rod, the force in the connecting rod will also vary. Further, the inclination of the crank is continually changing, and hence the turning effort on the crank pin, that is the force at right angles to the crank, is continually varying throughout each revolution. This turning effort is zero when the crank and the connecting rod are in the same straight line, that is when the engine is in its dead-centre positions.

The effect of the pressure on the piston in producing turning effort on the crank is shown diagrammatically in Fig. 2. If the pressure on the piston is p lb. per sq. in., then the total force on the piston is P lb., and $P = p \times A$, where A is the area of the piston in square inches. This force P is transmitted to the gudgeon pin and it induces a thrust, F, in the connecting rod, the magnitude of F depending upon the obliquity of the connecting rod, ϕ. An equal and an opposite force, F, is transmitted through the connecting rod to the crank pin, and this force can be shown to be equivalent to two forces, T and

R, the former acting perpendicularly to the crank and the latter along it, the magnitude of these forces depending upon θ, the inclination of the crank to the centre line of the engine. The force T is the turning effort which causes rotation of the crank shaft, while R produces a direct thrust on the crank shaft bearings. But in addition to the variation caused by the changes in the angles ϕ and θ, in the majority of reciprocating steam engines, the value of the force P is also changing during each stroke of the piston. As will be shown later,[1] it is far from being economical to admit steam at boiler pressure to the cylinder during the entire stroke of the piston, and so after the piston has moved through a fraction of its stroke, the supply is cut off, and for the remainder of the stroke the piston moves forward, due to the expansion of the steam. As expansion proceeds, the pressure falls, and thus the force on the piston also becomes less, which in turn affects the value of the forces F, T, and R. It is beyond the scope of these volumes to explain the methods whereby the turning effort during a complete revolution of the engine is determined, but the interested student will find the subject fully treated in the majority of textbooks on steam engines. It will be sufficient for our purpose to realise that the turning effort for a single-cylinder double-acting engine varies as shown in Fig. 3, where the figures along the base represent the values of θ, the inclination of the crank for one complete revolution, and the vertical ordinate represents the force T. If now from this diagram the value of T_m, the mean turning effort during the complete revolution, is found, and we draw a line at a distance T_m from the base, we see that the turning

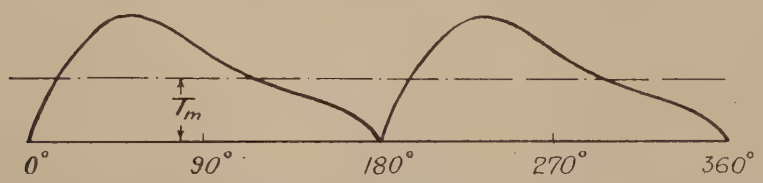


FIG. 3.—Turning Effort Diagram for One Revolution of the Crank of a Single-cylinder Double-acting Engine.

effort on the crank pin is first greater and then smaller than is necessary to maintain a constant speed. The flywheel is so designed that it absorbs the excess of energy developed in the cylinder when this is above the mean, and then gives it out again when the energy developed is below the mean, with only a very small specified variation in its speed of rotation and thus in the speed of rotation of the crank shaft of the engine.

When, however, only one cylinder is employed, although a flywheel of large diameter and of heavy weight may be fitted, it is impossible to keep the speed of rotation within very narrow limits. For driving certain classes of machinery, notably electrical generators developing alternating current under circumstances where two or more generators may have to be employed simultaneously for feeding the supply mains, it is very necessary that a constant speed of rotation should be obtained. Hence, instead of employing a single cylinder, two, three, or even four cylinders are employed, these driving on to separate cranks, all of which are connected so as to form one crank shaft. When two or four cylinders are employed, the cranks are generally connected together at right angles to each other, and in three-cylinder engines the cranks are arranged at 120° to each other. In this way a more even turning effort is obtained on the crank shaft, and thus, while employing a smaller flywheel, the speed variation during each revolution is kept within much closer limits. Fig. 4 shows the turning effort diagram for a three-cylinder double-acting engine with the cranks arranged at 120° to each other. The resultant turning effort on the crank shaft during one revolution is shown by the full line, while the straight line shows the mean turning effort. It will be observed that the fluctuation

[1] See page 10.

of turning effort on either side of the mean is very much less than it was in the case of the single-cylinder engine.

Another advantage which follows the use of the multi-crank engine is the facility with which it can be started up. It has already been noted that with a single-cylinder engine, on starting up, it is first of all necessary to bar the engine round to a suitable position before admitting steam to the cylinder. With large engines this may be a particularly laborious job, unless a small turning engine is provided for this special purpose. When two or more cylinders are employed, it is hardly possible that the engine would stop in such a position that on admitting steam the force on the pistons would not provide sufficient turning effort to set the engine in motion. Hence for engines which have to be frequently started and stopped, two or more cylinders are generally employed. Thus in a locomotive there are always two or more cylinders, while in the majority of reciprocating engines used for the propulsion of ships three cylinders are employed. It is obvious that in both of these instances the engine must be capable of being started up from any position. To avoid confusion, it may be mentioned here that in the tandem type of engine, in which one cylinder is arranged in line behind the other and the pistons are mounted on a common piston rod, we have a duplication of cylinders without a duplication of cranks. Here, the effect is to obtain a greater

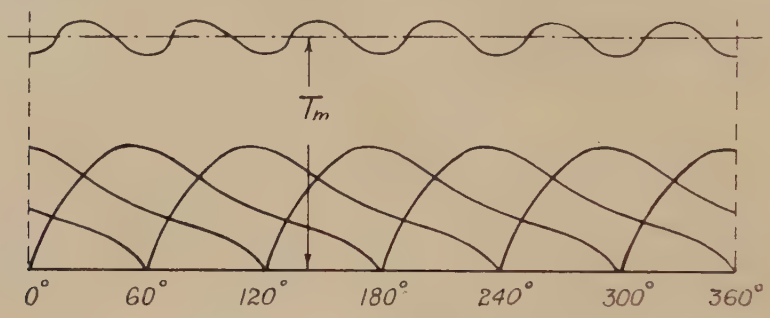


FIG. 4.—Turning Effort Diagram for a Three-cylinder Double-acting Engine.

power without having to employ cylinders of excessive dimensions. As we shall see when we come to describe some of the various modern forms of reciprocating steam engines, these tandem engines are often to be found with two lines of cylinders, that is with four cylinders, in which case the two cranks are at right angles to each other; where, however, only one line of cylinders is employed, some form of barring engine is usually installed.

But while the use of a heavy flywheel or of a multiplicity of cylinders in conjunction with a smaller flywheel will ensure a practically uniform speed during the entire revolution of the crank shaft, the sudden reversal of the motion of the pistons together with the piston rods, crossheads, and that part of the connecting rods that may be assumed to have a reciprocating motion, would always be accompanied by a thumping effect at the end of each stroke which would cause excessive wear on the gudgeon pin and crank shaft bearings; and as soon as these became the least bit slack, the noise would become excessive. To obviate this defect, the design of the valves controlling the admission of steam to the cylinders is such that they always open just before the piston reaches its dead centre. In this way a small quantity of steam is admitted to the cylinder and the piston in its backward motion is cushioned against this steam, being thus brought gradually to rest. This cushioning effect is found to be of very great value in producing a silent and smooth-running engine.

In the actual commercial operation of a reciprocating steam engine, or indeed of

any prime mover, it is not sufficient for the speed to be uniform during each revolution of the engine; it is generally of great importance that the speed should be kept at the same value during the entire period that the engine is running. Thus when driving electrical generators, no matter whether the generator is being called upon to supply current up to its full capacity or whether it is running on very light load, the speed of rotation of the generator must be maintained at a constant figure in order that the voltage of the electrical supply may be kept uniform. Again, when driving weaving looms it is of considerable importance that a uniform speed should be maintained all the time the looms are working, despite the number of looms actually being driven by the engine. In order to maintain a constant engine speed despite the fluctuations in the power which it is called upon to develop, a device termed a "governor" is fitted. There are a number of different forms of governor in use at the present day, but the principle upon which they all work is substantially the same. It is well known that a heavy weight rotating in a circle has a tendency to fly away from the centre of the circle. The simple experiment with a stone attached to a piece of string and whirled around in a circle yields a ready proof of the existence of this force. This force is known as centrifugal force, and because the majority of governors make use of this force they are generally known as centrifugal governors. The tendency which two or more

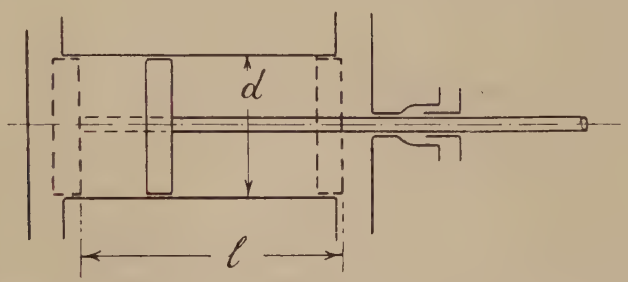


FIG. 5.—Diagram of Cylinder of Engine to show work done by Expansion of Steam.

revolving weights have to fly outward when the speed of rotation is increased is employed to move a sleeve on the governor spindle, whence by a suitable mechanism or by the use of oil pressure the movement of the sleeve is transmitted either to a throttle on the main steam supply or to the admission valves on the engine. In the first case the pressure of the steam supplied to the engine is varied, and in the second the time during which steam is admitted to the cylinder during each piston stroke is varied. In either case, when the speed has a tendency to fall below the normal, the power developed by the engine is increased, and when the speed tends to rise, the power developed by the engine is decreased. Thus, with a sensitive governor, despite the fluctuation of the load on the engine over a very wide range from no load even to a considerable overload, the speed of the engine remains practically constant. Hence, it will be realised that the difference between the flywheel and the governor is fundamental, for while the former merely maintains a constant speed during each revolution of the engine, the governor keeps the engine running at a constant speed during the whole period that the engine is in operation, despite the variation in the load. Or, expressed in other words, the flywheel tends to eliminate cyclic variations in speed, while the governor eliminates variations of speed due to changes in the external conditions affecting the load on the engine.

Mention has already been made of the fact that in the majority of engines the steam entering the cylinder is cut off after the piston has traversed a portion of its stroke, and that for the remainder, the piston moves under the influence of the expansion of the steam. We must now pause for a moment to investigate the reasons for using the steam expansively. Consider the cylinder of an engine as shown in Fig. 5. The diameter of

the piston, d, we will assume to be 12 in., and the stroke of the piston, l, we will take as 24 in., or 2 ft. For the sake of simplicity we will further assume that the pressure of the steam entering the cylinder is 200 lb. per sq. in. Imagine now that the steam is admitted during the entire stroke of the piston. The force on the piston is equal to pressure × area, and the work done is equal to force × distance.[1] Thus we have—

$$\text{Area of piston} = \frac{\pi d^2}{4} = \frac{\pi \times 12^2}{4} = 113 \text{ sq. in.}$$
$$\text{Force on piston} = 200 \times 113 = 22{,}600 \text{ lb.}$$
$$\text{Work done per stroke} = 22{,}600 \times 2 = 45{,}200 \text{ ft. lb.}$$

Or, since Work done = Force on piston × stroke, and Force on piston = Pressure × area of piston, Work done = Pressure × area × stroke = Pressure × volume, since Area of piston × stroke = Volume.

We desire next to ascertain the weight of steam utilised in performing this amount of work. In order to determine various facts regarding steam at different pressures, steam tables have been prepared in which the information most frequently required is given. A copy of the steam tables as usually employed by engineers is given at the end of Volume IV of this work. From these tables we find that at a pressure of 200 lb. per sq. in., 1 lb. of steam occupies a volume of 2·12 cu. ft., or in other words 1 cu. ft. weighs $\frac{1}{2{\cdot}12}$ lb. The volume of steam admitted to the cylinder is $\frac{\pi d^2}{4} \times l = \frac{\pi \times 1^2}{4} \times 2 = 1{\cdot}57$ cu. ft. (both d and l in this case are expressed in feet);

$$\therefore \text{Weight of steam used} = \frac{1{\cdot}57}{2{\cdot}12} = 0{\cdot}741 \text{ lb.}$$

But this weight of steam is responsible for the 45,200 ft. lb. of work done.

$$\text{Hence, Work done per lb. of steam} = \frac{45{,}200}{0{\cdot}741} = 61{,}000 \text{ ft. lb.}$$

Imagine now that the steam, instead of being admitted to the cylinder for the entire stroke of the piston, is cut off when the piston has just described one-half of its stroke, and that for the remainder of its stroke the piston is driven forward by the expansion of the steam. It is apparent that the weight of steam used will now be only one-half of what it was previously, i.e. $\frac{0{\cdot}741}{2} = 0{\cdot}370$ lb. In order to determine the work done by this steam during the complete stroke of the piston, we may consider (1) the work done during the admission of steam, and (2) work done during the expansion of the steam, the amount of work represented by (2) being, of course, obtained without the consumption of any additional steam from the boiler. When steam expands in a cylinder the assumption is made that it expands in such a way that the product of the pressure and the volume is constant, i.e. the law governing the expansion is expressed by the relationship $P \times V = k$, where k is a constant. This is known as Boyle's Law. If now we plot a curve such that the distances measured horizontally represent changes in volume and vertical heights represent changes of pressure, the curve obtained when a steam expands in accordance with Boyle's Law is called a "rectangular hyperbola." Such a curve for steam in the cylinder of an engine with cut-off at half stroke is shown in Fig. 6. Further, it can be shown that the area of such a diagram is a measure of the work done by the steam. Thus we have that the work done during admission is

[1] In calculating the work done, we have to remember that the unit of work is the foot-pound. Hence while Force on piston = Pressure in lb. on each square inch × area in sq. in., Work done = Force in lb. × distance in ft.

represented by the area ABEF and the work done during expansion by the area BCDE. Further, the work done during expansion can be calculated from the expression—

Work done during expansion = PV $\log_e$ R
where P = Pressure of steam at beginning of expansion.
V = Volume of steam at beginning of expansion.
R = Ratio of volume of steam at end of expansion to volume at beginning of expansion.

R is briefly termed the "ratio of expansion" and $\log_e$ R is the hyperbolic logarithm of R.[1]

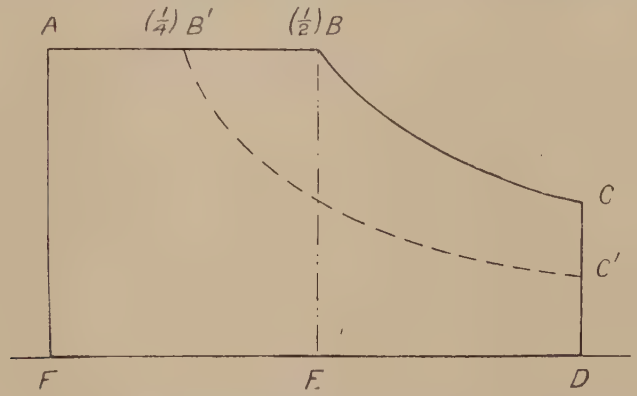


FIG. 6.—Theoretical Indicator Diagram to show the Effect of Expansion of Steam.

Returning now to our calculation of the work done when cut-off occurs at half stroke, we have:

Work done during admission = Force on piston × distance = 22,600 × 1 = 22,600 ft. lb.; or alternatively Work done during admission = Pressure × Volume of steam at cut-off = PV, where P is expressed in lb. per sq. ft. and V in cu. ft.

Also Work done during expansion = PV $\log_e$ R.

Here R = 2 and $\log_e$ 2 = 0·693, and since we desire to express the work done in foot-pounds, P must be expressed in lb. per sq. ft. and V in cu. ft.

$$\therefore \quad \text{PV} \log_e \text{R} = (200 \times 144) \frac{1{\cdot}57}{2} \times 0{\cdot}693$$
$$= 15{,}670 \text{ ft. lb.}$$

Hence, Total work done on piston
$$= 22{,}600 + 15{,}670$$
$$= 38{,}270 \text{ ft. lb.}$$

But weight of steam used $= 0{\cdot}370$ lb.

$$\therefore \text{ Work done per lb. of steam} = \frac{38{,}270}{0{\cdot}370} = 103{,}300 \text{ ft. lb.}$$

It will be seen, therefore, in comparison with the results obtained when the steam was admitted for the entire stroke, that when steam is cut off at half stroke, the work done in the cylinder is somewhat less, but the work done per lb. of steam is considerably greater.

Consider next the same engine but with steam cut off at quarter stroke. The

[1] In many engineering calculations logarithms, or briefly logs., are employed, and a table of logs. is given at the end of Volume IV of this work. These, however, are calculated to the base 10. To find logs. to the base e, we have the simple rule:

$$\log_e x = \log_{10} x \times 2{\cdot}3026.$$

expansion now follows the line B′C′ in Fig. 6. In this case $R = 4$ and from the tables $\log_e 4 = 1{\cdot}386$.

Then, Work done during admission = Force on piston × distance

$$= 22{,}600 \times \tfrac{1}{2} = 11{,}300 \text{ ft. lb.}$$

Work done during expansion $= PV \log_e R$

$$= (200 \times 144)\frac{1{\cdot}57}{4} = 1{\cdot}386$$

$$= 15{,}670 \text{ ft. lb.}$$

Thus, Total work done on piston $= 11{,}300 + 15{,}670$

$$= 26{,}970 \text{ ft. lb.}$$

But the weight of steam admitted to the cylinder is now only $\frac{0{\cdot}741}{4} = 0{\cdot}185$ lb.

$\therefore$ Work done per lb. of steam $= \frac{26{,}970}{0{\cdot}185} = 145{,}000$ ft. lb.

Again it will be seen in comparison with the results obtained above, that while the work done during the entire stroke of the piston is reduced, the work done per pound of steam is increased.

Similarly, we can determine the amount of work done per stroke and the work done per lb. of steam as the cut-off becomes earlier and earlier, say, at one-eighth and one-sixteenth stroke. It is not necessary to give the details of the calculation here, but the results obtained together with those found above can be tabulated as follows:

Point of Cut-off.	R.	$\log_e$ R.	Volume of Steam at Cut-off (V).	Weight of Steam at Cut-off.	Work Done during Admission = PV.	Work Done during Expansion = PV $\log_e$ R.	Total Work Done per Stroke.	Work Done per Pound of Steam.
			cu. ft.	lb.	ft. lb.	ft. lb.	ft. lb.	ft. lb.
Full Stroke .	—	—	1·570	0·741	45,200	—	45,200	61,000
½ Stroke .	2	0·693	0·785	0·370	22,600	15,670	38,270	103,300
¼ Stroke .	4	1·386	0·392	0·185	11,300	15,670	26,970	145,500
⅛ Stroke .	8	2·079	0·196	0·092	5,650	11,730	17,380	188,800
1/16 Stroke .	16	2·772	0·098	0·046	2,825	7,835	10,660	231,800

It is apparent, from the figures given in the above table, that as the cut-off is made earlier and earlier the amount of work obtained from each pound of steam becomes greater and greater, although the total amount of work done during each complete stroke of the piston is less and less. Or to look at it from a different standpoint, if we have two similar cylinders supplied with steam at the same pressure, but with one receiving steam for the entire stroke of the piston, while the other receives steam for only one-sixteenth of the stroke after which the steam expands, driving the piston forward for the remainder of the stroke, then the total amount of work done in the former case will be nearly four times as much as in the latter; but against this the work done per pound of steam in the former engine will be only one-quarter of what it is in the latter. It will be seen, therefore, that it is a highly economical procedure to use the steam expansively in an engine, since by so doing, the work done by a given weight of the steam, and therefore by a given weight of fuel used in the boiler or boilers producing that steam, is very much greater than if the steam is not allowed to expand. The cost of fuel is by far the largest item to be met in the running of an engine, and hence the greater the degree of expansion permitted, the cheaper will be the cost of generating the power

There is, however, a limit to the amount of expansion that can be economically permitted, and we must now endeavour to explain briefly the reason for this.

When an engine is required to develop a certain power, the size of the cylinder and therefore the overall dimensions and the weight must be kept as small as possible, because the larger the engine the greater its initial cost and the larger will be the ground space it occupies in the power house or the works. When an excessive degree of expansion is permitted, the size of the cylinder and with it the size of all the other parts of the engine become excessive, and the designer is thus forced to compromise between his desire to produce an economical engine and the necessity of keeping his engine within moderate dimensions. Another reason which tends to limit the amount of expansion which is permitted to occur in the cylinder should be noted. When the steam expands, the pressure falls ; and this fall of pressure is accompanied by a fall in temperature. The greater the degree of expansion, the greater will be the fall in pressure and therefore in temperature. Thus when the steam has completed its expansion and the exhaust ports are opened, the temperature of the steam, and with it the temperature of the cylinder walls, is very much lower than that of the steam at boiler pressure. Some idea of the fall of temperature can be gathered from a study of the steam tables. Thus at a pressure of 200 lb. per sq. in. gauge, the temperature of the steam is 388° F., and if this steam is permitted to expand down to a pressure of 20 lb. per sq. in., the temperature falls to 228° F. Imagine now that steam is permitted to expand through this range of pressure and that after expansion the steam passes through the exhaust port. After the piston has moved back to the beginning of the stroke, steam at boiler pressure is again admitted. This steam comes into contact with the walls of the cylinder, which have been cooled by contact with the low-temperature steam resulting from the previous expansion. As a consequence of this, a considerable amount of steam is condensed before the temperature of the cylinder walls is again raised to that of the incoming steam, and it is obvious that the greater the degree of expansion occurring in the cylinder the greater will be the loss due to condensation of the incoming steam at the beginning of each stroke.

Apart from the actual loss of efficiency due to the initial condensation of steam—for steam which is thus condensed cannot do any useful work in the cylinder—the presence of water in the cylinder may have positively harmful effects on the running of the engine. The moisture formed on the cylinder walls will gradually trickle down and accumulate at the bottom of the cylinder, and if a sufficiently large volume of it accumulates in the cylinder, it may be driven by the piston against the cylinder cover, causing a knocking effect known as water hammer. But water is practically an incompressible fluid and hence the water hammer may occur with such force as to cause fracture. Relief valves are always fitted in the cylinder which lift when the pressure tends to rise above the normal, but these may become inoperative from some cause or another, giving rise to water hammer, accompanied possibly by disastrous consequences. Many accidents have occurred as a result of water accumulating in the cylinders or passages of a steam engine. It was in the past common practice with large engines to construct a jacket around the cylinder, this jacket being supplied with steam at boiler pressure, thus tending to keep the cylinder walls hot, while to-day in all up-to-date steam engine plant the steam is superheated before it enters the engine, and in these ways the amount of condensation may be reduced. But no matter what steps are taken to eliminate the evils of cylinder condensation, they can only be effectively reduced when the range of temperature in the cylinder is limited, by permitting only a small degree of expansion. It is difficult to say exactly what is considered the most suitable point at which cut-off should occur, since different makers of reciprocating steam engines have different ideas on the subject ; but as we shall see when we come to consider the various types now on the market, cut-off at about one-tenth of the stroke can be regarded as yielding a suitable compromise in those engines where the entire expansion takes place in one cylinder.

With a view to taking the fullest advantage of the economy which results from expanding the steam down to the lowest pressure that can be obtained and at the same time to reduce the range of temperature occurring in one cylinder, "compounding" is frequently adopted. Compounding consists of using two, three, or even four cylinders in conjunction with each other in such a way that the steam, after doing work in the first cylinder, passes to the second, where it again expands, doing work, and from which it exhausts perhaps to a third cylinder or even a fourth, expanding in each before it is permitted to escape. As the steam expands, its volume increases, and hence the cylinders have to be made successively larger in order to cope with the increasing volume of steam passing through them. When two cylinders are employed, the steam expanding first

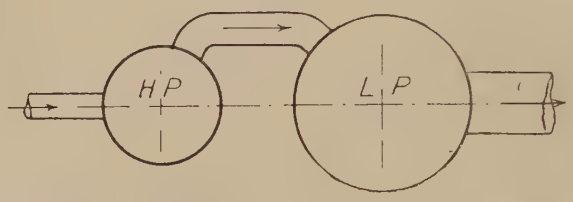


FIG. 7.—Arrangement of Cylinders and Steam Pipes in a Two-crank Compound Engine.

in one cylinder and then in the second, we have what is generally termed a compound engine. The arrangement of the cylinders in such an engine is shown diagrammatically in Fig. 7. These engines are made sometimes with the axes of the cylinders parallel to each other, there being thus two piston rods, two connecting rods, and two cranks, and sometimes with the two cylinders having a common axis, there being thus only one piston rod, one connecting rod, and one crank. In the latter case the engine is called a tandem compound engine, the arrangement of cylinders being as shown in Fig. 8. In both cases, the cylinder which receives steam direct from the boiler is called the high-pressure (h.p.) cylinder and the other is called the low-pressure (l.p.) cylinder.

When three cylinders are employed, the steam expanding through each, we have

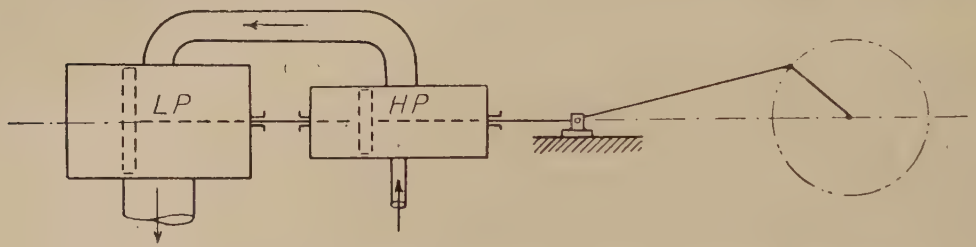


FIG. 8.—Diagrammatic Arrangement of Horizontal Tandem Compound Steam Engine.

what is termed a "triple-expansion" engine, and the arrangement of cylinders is as shown diagrammatically in Fig. 9. It may be mentioned that this type of engine is sometimes designed with the cylinders arranged on a common axis, although it will far more frequently be found with the axes of the cylinders parallel to each other, there being thus three cranks. The cylinders are referred to as the high-pressure (h.p.), the intermediate-pressure (i.p.), and the low-pressure (l.p.) in the order in which they receive the steam.

Where four cylinders are employed, we have what is termed the quadruple-expansion engine; the cylinders are arranged with their axes parallel to each other and the pistons drive four cranks. The cylinders, in the order in which they receive the steam, are termed the high-pressure, first intermediate, second intermediate, and low-pressure cylinders respectively. In these days when steam turbines are generally employed

for the generation of large powers, the quadruple-expansion engine is seldom made, except perhaps for the propulsion of ships. There is, however, an interesting type of four-cylinder engine which is frequently met with, particularly in marine practice, and which might easily be mistaken for a quadruple-expansion engine. This is the four-cylinder triple-expansion engine, the arrangement of the cylinders being as shown in Fig. 10. For engines of large power, the size of the cylinders have to be correspondingly large. It follows, therefore, that where a large high-pressure cylinder is required, the intermediate-pressure cylinder will be larger and the low-pressure cylinder may become of excessive dimensions. To obviate this difficulty, two low-pressure cylinders are

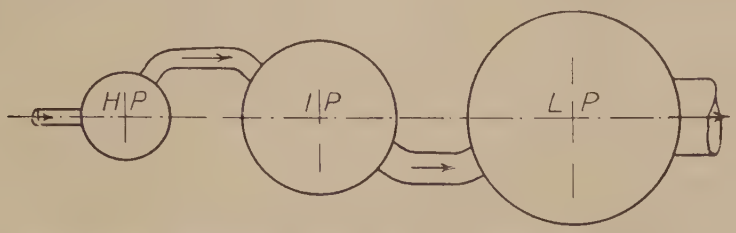


FIG. 9.—Arrangement of Cylinders and Steam Pipes for Three-crank Triple-expansion Engine.

employed of such a diameter that the total volume of the two is equal to the volume of the single large l.p. cylinder that would otherwise be necessary. The steam from the i.p. cylinder passes into two pipes, each of which leads to one of the l.p. cylinders, and in marine work the low-pressure cylinders are referred to as the forward low-pressure (f.l.p.) cylinder and the after low-pressure (a.l.p.) cylinder, according to their position in the ship. Another reason why this particular arrangement of cylinders is preferred to the simpler triple-expansion arrangement lies in the fact that it enables a better-balanced engine to be designed. While the three-crank engine can be balanced sufficiently well for all ordinary purposes, a perfect balance cannot be obtained. With four cranks, however, the balance can be made scientifically perfect, and hence for high-

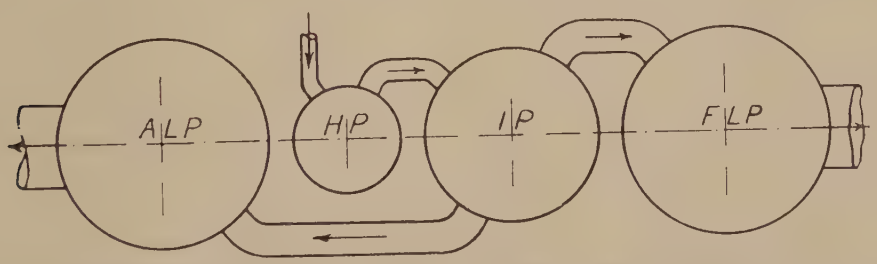


FIG. 10.—Arrangement of Cylinders and Steam Pipes of Four-cylinder Triple-expansion Engine.

speed passenger liners, where the speed of rotation of the propellers and therefore of the engines driving them is higher than on cargo vessels, and where, moreover, a smoother-running engine with freedom from vibration is necessary, the four-cylinder, triple-expansion engine is often adopted. The cranks are not arranged at 90° to each other, as might be expected, but in accordance with a system which has been developed in order to obtain perfect balance and which is known as the Yarrow, Schlick and Tweedy system of balancing.

When steam is permitted to expand in two or more cylinders, the disadvantages which follow the employment of a high degree of expansion are to a very large extent eliminated, and hence, with a view to obtaining the fullest advantage of the economy which, as we have seen, results from expansion, the designers of steam engines arrange

for the expansion to be carried out to the furthest possible limit. It might perhaps be thought that if the steam is allowed to expand from boiler pressure down to a pressure of 2 or 3 lb. per sq. in. above atmospheric pressure, this would be the utmost limit to which the expansion could be carried. The small pressure 2 or 3 lb. per sq. in. above atmospheric is, of course, absolutely necessary if the steam at the end of expansion is to possess sufficient energy to carry it through the exhaust passages, however liberally and smoothly these may have been designed. It must not be forgotten, however, that the atmosphere itself is exerting a pressure on every surface with which it is in contact, and this pressure, which can be readily measured by the barometer, is about 14·7 lb. per sq. in., or as it would be shown on the barometer it is equivalent to the weight of a column of mercury 30 in. high. Suppose then it were possible to create a more or less perfect vacuum at the exhaust end of our low-pressure cylinder, we should be able to permit the steam to expand to a considerably lower pressure than if it were compelled to exhaust into the atmosphere. But such a vacuum can be easily created by the simple expedient of condensing the steam after it leaves the cylinder, since when steam is converted into water, that is when it is condensed, its volume is diminished very considerably, and if no air is allowed to leak into the vessel in which condensation takes place a very considerable lowering of the pressure will result. Hence we find that in the majority

Fig. 11.—Theoretical Indicator Diagram showing work done by a Condensing Steam Engine.

of steam engines, instead of allowing the steam to expand down to about atmospheric pressure, a condenser is fitted and the exhaust from the low-pressure cylinder passes into this condenser, where it is converted into water, with the result that the pressure at the exhaust is maintained at a very low figure indeed.

Some idea of the enormous economy which follows the use of condensing apparatus can be gathered from Fig. 11, which shows approximately to scale the extra work which is obtained when steam is further expanded from a pressure of 2 lb. per sq. in. above atmospheric down to a vacuum of only 2 lb. per sq. in. This latter pressure, it may be noted, is generally referred to as 2 lb. per sq. in. absolute, i.e. 2 lb. per sq. in. above zero pressure, to distinguish it from the pressure of 2 lb. per sq. in. above atmospheric. The pressure gauges on an engine usually point to zero at atmospheric pressure, so the latter figure is often referred to as 2 lb. per sq. in. gauge, to distinguish it from absolute pressure. Actually the absolute pressure is 14·7 + 2 = 16·7 lb. per sq. in. absolute, and to give us even figures to deal with we can usually take the pressure of the atmosphere as 15 lb. per sq. in., and we have the important rule :

Absolute pressure = Gauge pressure + 15.

It should be noted that pressures in the steam tables are given in absolute units.

To return to Fig. 11. Steam at a pressure of 215 lb. per sq. in. absolute (200 lb. per sq. in. gauge) is first of all expanded down to 17 lb. per sq. in. absolute, this being

represented by the line AB. It can be shown that the work done is represented by the area ABCD. If, however, a condenser is fitted and the steam is expanded down to a pressure of 2 lb. per sq. in. absolute, the expansion follows the line ABE and the total work done is represented by the area AEFD. Thus the shaded area BEFC represents the extra work which is obtained from the steam without any increase in the expenditure of fuel in generating that steam. From the steam tables we see that at a pressure of 2 lb. per sq. in. absolute the volume occupied by 1 lb. of steam is approximately 172 cu. ft. Now, 1 cu. ft. of water weighs 62·3 lb., so that 1 lb. of water occupies 0·016 cu. ft., hence on condensation occurring 172 cu. ft. of steam is reduced to 0·016 cu. ft. of water, or the volume is reduced to less than one-ten-thousandth part of what it was formerly. Other factors have to be taken into account,[1] but the above figures are sufficiently accurate to indicate why condensation results in the production of a vacuum.

There is a more obvious reason why steps should be taken to condense the steam after it has done its work in the engine, apart altogether from the extra degree of expansion which is thus rendered possible, and that is because it enables the water required for the production of the steam to be used over and over again. After the steam has passed into the condenser, it is converted into water, from whence it is pumped into a tank known as the hot well, and from this tank it is again pumped into the boiler as required. It may be necessary to filter the water before it is again sent into the boiler, as during its passage through the engine it may pick up small globules of oil from the lubricated surfaces with which it comes into contact, and oil in the boiler, even in minute traces, clings to the surfaces of the plates and tubes, reducing their conductivity and thus leading to overheating and corrosion. If it is to be suitable for pumping into a boiler, water has to be very pure, in fact drinking water is not generally considered sufficiently pure for boiler-feed purposes, as it contains substances in solution which, while not injurious to the health of human beings, are injurious to a boiler, since they become deposited on the heating surfaces as fur or scale, with somewhat similar effects as the oil referred to above. Water which is produced by the condensation of steam is free from these dissolved impurities. Moreover, in the majority of cases the water which is used in a power house has to be paid for, and hence for this reason as well as because of the evil effects of scale it is highly advantageous to employ condensing plant. But even where pure water is available at little or no cost, as, for example, when a well has been sunk on the property upon which the engine is installed or where water-purifying plant is employed, it will generally be found that a condenser is fitted, so that the economy of a greater degree of expansion can be obtained, despite the extra initial cost which the condenser and its necessary pumps, etc., involves.

In the case of steam engines used for the propulsion of ships, condensation is absolutely essential. It would be almost a matter of impossibility for a ship to carry a sufficiently large supply of fresh water to supply the boilers for a long voyage, say from this country to the United States or Canada, and even if it were possible, the weight of the water and the space it would occupy would not permit of the carriage of very much else in the way of passengers or cargo. At the same time it would be of little use to pump sea water into the boilers, since after a short while the interior surfaces of the boilers would be so encrusted with sea salt that the heat from the furnaces would only with great difficulty reach the water, and an enormous expenditure of fuel would be necessary to generate sufficient steam for the engines. It might almost be said that what was saved by not carrying fresh water was lost again in the extra fuel which would be necessary. Hence we find that on board ship particular attention is paid to the condensing apparatus with a view to preventing even the smallest loss of steam during

[1] The question of condensing is discussed at greater length in Chapters VIII and IX, Volume I.

the voyage. The feed water tanks are filled with fresh water when the vessel is in port, and the boilers are also filled to the working level, but when the vessel is at sea and the engines are working, the condensed steam, or condensate as it is termed, is filtered and returned to the boilers over and over again, and by paying prompt attention to any possible source of loss of steam or of water, the supply is generally sufficient for the entire voyage. If undue loss occurs and sufficient fresh water is not available in the tanks, or if it is desired to conserve the water carried in the tanks, there is generally an evaporator installed in the engine room which produces steam from sea water, the steam, which is of course pure, being passed direct to the condenser, where it mixes with the exhaust and ultimately finds its way as pure water into the boilers.

There is one class of steam engine, and a very important one, to which condensers are not fitted, and this is the locomotive. Here, because of the peculiar conditions governing the design and operation of the locomotive, it is not considered practicable to fit a condenser. We have already described two different types of locomotive to which condensers have been fitted,[1] but these are quite exceptional designs. In the ordinary steam locomotive the carrying of a condenser together with the necessary pumps and the cooling water would involve so great an increase of weight and would take up so much space that it is hardly considered feasible. Thus the steam locomotive is compelled to exhaust into the atmosphere, and, from the point of view of power developed per ton of coal burned, it is therefore one of the most inefficient prime movers that is used on a commercial scale. Further, it is necessary to carry sufficient water either in tanks mounted on the engine frame or on a separate vehicle called a tender attached to the engine, and even then on some of our main lines, troughs have been laid down so that the water in the tender can be replenished when necessary. Many locomotive engineers have considered the possibility of adopting some system of condensation, but except in a few experimental locomotives it has not been considered commercially practicable. The reader will not, however, imagine that British-built locomotives are otherwise inefficient. Except that condensing with its attendant economics is not deemed advisable, it can be stated without fear of contradiction that a modern express locomotive is one of the finest examples of a self-contained, high-powered prime mover that it is possible to construct.

We have endeavoured in the foregoing pages of this chapter to set forth in simple language the basic principles underlying the design and operation of the reciprocating steam engine. It will now be our endeavour in the succeeding chapters to give some examples of modern reciprocating engines of different types. We must point out, however, that there are so very many different types of reciprocating steam engine that it is obviously impossible with the space at our command to include more than a few of the most representative types. But if the reader has thoroughly understood the principles set forth above, and if he studies carefully the examples which are described in the subsequent chapters, he should have no difficulty in understanding the peculiarities of any other type with which he may come into contact.

[1] See Volume I, pages 14 and 30.

CHAPTER II

THE VERTICAL RECIPROCATING STEAM ENGINE—SOME MODERN EXAMPLES

WHEN one considers the number of years during which the reciprocating steam engine has been in use and the very large number of different uses to which it is put, it is easy to realise that there are very many different types of this engine upon the market. Nor, indeed, does the number show any sign of diminishing. We have seen in the first volume of this work that the steam turbine has tended largely to displace the reciprocating steam engine for a number of purposes both on land and afloat, but this has only led the manufacturers of the latter type to improve their designs and in many cases to evolve entirely new designs, so that it cannot yet be said that finality has been reached. The vast number of different types of steam reciprocating engines makes it a difficult matter to decide which designs can be most suitably included in a work of this nature, and in view of this difficulty it has been considered advisable to deal first of all with stationary engines, and thence to deal with the moving types, i.e. the types which are employed for driving locomotives, lorries, and ships, and as far as possible we will first describe units of small power, proceeding thence to the units of large output.

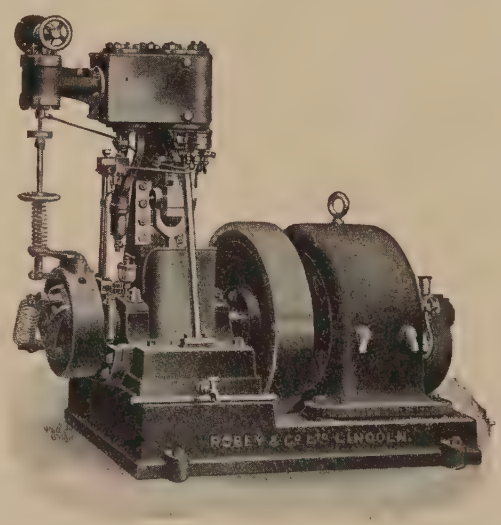


FIG. 12.—Single-cylinder, High-speed, Open-type Vertical Engine by Messrs. Robey & Co., Ltd., Lincoln.

The small reciprocating steam engine is very extensively adopted at the present time for driving dynamos, pumps, compressors, etc., owing to the fact that the steam turbine with few exceptions cannot be economically employed for the generation of small powers. These small reciprocating engines are generally of the high-speed vertical type, and in their simplest form have merely one cylinder, the steam from the main stop valve doing work on the piston from whence it passes direct to the condenser or to the atmosphere. A very good example of an engine of this type is shown in Fig. 12, which is a reproduction from a photograph of a single-cylinder, high-speed, vertical, open-type engine direct coupled to an electrical generator. This engine is designed and constructed by Messrs. Robey & Co., Ltd., at their Globe Works, Lincoln, and is built in various sizes from 2 h.p. up to about 140 h.p., according to the dimensions of the cylinder. Thus considering a unit of medium size having a cylinder $7\frac{1}{2}$ in. diam. with piston stroke 8 in., if the pressure of the steam at the stop

valve is 150 lb. per sq. in., the engine will have a speed of 400 revs. per min. and will develop 32 effective brake horse power, or in other words it could drive a dynamo of

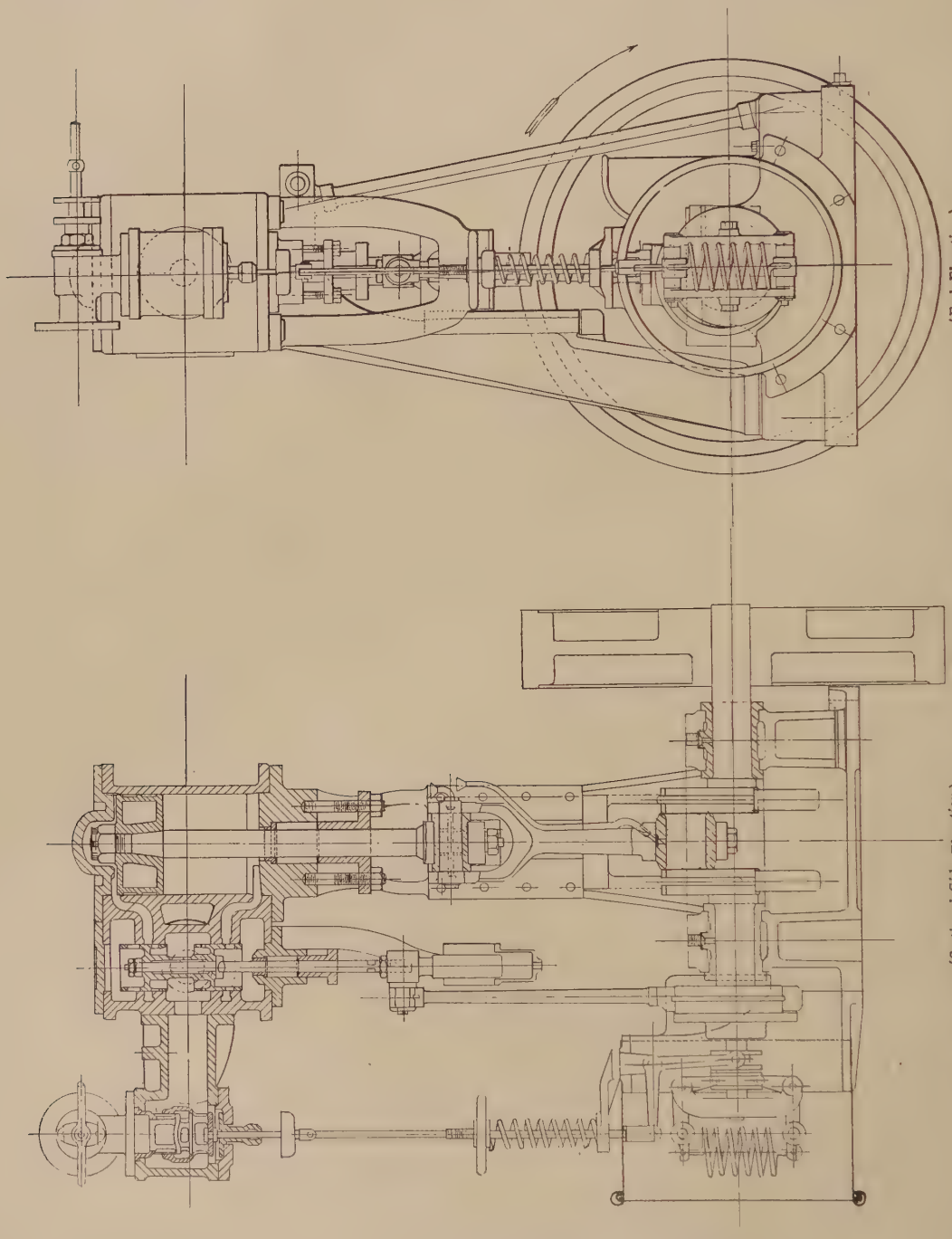


FIG. 13.—General Arrangement Drawings of Robey Single-cylinder, High-speed, Open-type Vertical Engine. (*End Elevation.*) (*Sectional Side Elevation.*)

about 22½ kw. output. The smallest units have a cylinder 3½ in. diam. by 3 in. stroke, while the largest have a cylinder 14½ in. by 14 in.

A very good idea of the details of the design of these engines can be gathered from

Fig. 13, which shows a sectional side elevation and an end elevation, the dynamo being removed. It will be seen that the cylinder and the valve chest form a single casting, this being made of close-grain cast iron. The lower cylinder cover and the lower valve chest cover are fitted with stuffing boxes, both the neck bushes and the glands being made of gun-metal. The piston is also made of cast iron fitted with expanding cast-iron rings to ensure steam-tightness. The piston rod is of steel coned at the top end to take the piston and forged solid at the bottom to form the crosshead. The crosshead is fitted with a single cast-iron slipper to take the thrust due to the obliquity of the connecting rod, this slipper being provided with suitable means of adjustment for wear. The connecting rod is of the forked type with the gudgeon pin shrunk in, the middle portion of the pin being carried in an adjustable bearing. The bottom end of the connecting rod, or the " big end " as it is often termed, is of the marine type—that is, it carries the crank pin brasses, which are of necessity divided through the centre, and is fitted with adjustable liners, the brasses being held in position by two bolts which pass through a steel plate under the brasses, the brasses themselves, and through the bottom end of the connecting rod. These bolts are fitted with lock nuts to prevent the possibility of their working slack. The crank shaft is of steel and is supported in two

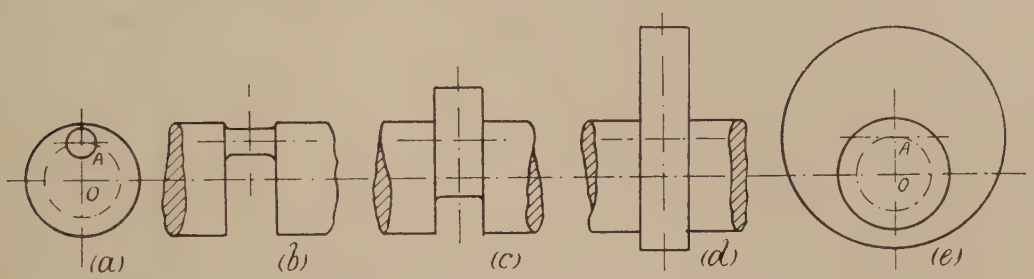


Fig. 14.—Diagram showing the Function of an Eccentric.

bearings, one on either side of the crank webs. A projection of the shaft to the right carries the flywheel, the boss of the flywheel forming one portion of the coupling for the dynamo or whatever it may be the engine is coupled to. The other end of the crank shaft carries the eccentric sheave by means of which the valve is driven, while at the end of the shaft the governor is mounted.

It will be of interest perhaps to pause for a moment in order to explain more fully the function of the eccentric. An eccentric is a circular disc of metal which is mounted on a shaft in such a way that the centre of the disc does not coincide with the centre of the shaft—that is, the disc is mounted eccentrically, and hence the name given to this device. Thus if in Fig. 14 (*e*), O is the centre of the shaft and A the centre of the eccentric, OA is the eccentricity of the eccentric, and it is obvious that as the shaft rotates A will describe a circle about O as centre as shown. If instead of the disc we had a small pin at A formed by cutting away the material of the shaft as shown in Fig. 14 (*a*) and (*b*), we can consider this arrangement as forming a little crank, and it is apparent that if the valve is driven by a rod attached to this pin, as the shaft rotates the valve will have a travel equal to 2 × OA. But as shown at (*b*), the shaft will be considerably weakened and the strength of the pin itself will also be small, and hence, in order to add to the strength, we can make the pin of greater diameter as at (*c*) without altering the distance OA. The travel of the valve will not, however, be affected in any way by increasing the diameter of the pin, so that if we still further increase its diameter until we get the arrangement shown at (*d*), we do not have to cut the shaft at all. Thus we get an arrangement which gives a very small travel to

the valve as the shaft rotates, the travel being always twice the eccentricity. The eccentric rod is secured to the valve rod by means of a simple form of gudgeon pin, and at its lower end it carries the eccentric strap, which is virtually a bearing of very large diameter.

Returning now to our study of Fig. 13, the eccentric can be seen in the sectional elevation to the left of the left-hand main bearing with the eccentric rod leading up to the valve rod. It is beyond the scope of this book to deal with the problems of the design and setting of eccentrics, but it may be mentioned briefly that the eccentricity of the eccentric must be such as will give an adequate travel to the valve, while it must be mounted on the shaft in such a position relatively to the crank that it gives proper distribution of the steam first to one side and then to the other side of the piston, giving a correct point of cut-off so that the steam can work expansively in the cylinder, and finally, when the piston has practically reached the end of its stroke, allowing the steam to pass out of the cylinder to the exhaust pipe. The reader who is interested in this subject will find the problem of setting the slide valve, which of course includes the eccentric, treated fully in textbooks on the steam engine. In order to relieve the valve chest stuffing box of any side pressure due to the slight obliquity of the eccentric rod, the valve rod works in a substantial guide which is supported from the underside of the cylinder casting as shown. The slide valve itself, which is of the piston type, is secured to the rod by means of an adjustable cotter and a locking nut, so that exact adjustment of the position of the valve relatively to the steam and exhaust ports can be made. Steam is admitted to the engine through the stop valve seen on the left in Fig. 12, this valve being mounted over the governor valve, from whence the steam passes into the valve chest, being admitted to the cylinder through the ports from the inside of the valve. After doing its work, the steam passes along the same ports to the outside of the valve and thence to the exhaust pipe, the flange of this pipe being shown dotted in Fig. 13.

The governor, which, as already mentioned, is mounted on the end of the crank shaft, is of the centrifugal type, comprising two heavy weights held together by a powerful spring. These weights are mounted on bell crank levers, and as the weights move inwards or outwards, due to a decrease or an increase in the speed of rotation of the shaft, they cause the other ends of the bell crank levers to move in a direction parallel to the shaft, thus sliding a collar along the shaft. This collar is grooved, and in this groove is fitted the ends of another bell-crank lever, so that as the rotating collar slides along the shaft it causes the bell-crank lever to swing about its fulcrum, causing a small vertical motion at the other end of the lever. This motion is transmitted through a spring-controlled vertical rod to a throttle valve, which is set in such a position that when the engine is running at normal speed under normal load, the requisite quantity of steam can pass. If the load increases, the speed has a tendency to drop, the governor weights move inwards, and this motion is transmitted to the throttle valve, which opens farther, thus permitting a greater quantity of steam to enter the cylinder and causing a rise of pressure in the valve chest. If, however, the load decreases, the reverse action takes place, the opening of the throttle valve becoming smaller. This type of governor is termed a " throttle " governor, to distinguish it from the " cut-off " or " variable-expansion " governor which is often fitted to reciprocating steam engines and in which, as the name implies, the supply of steam to the cylinder is cut off later in the stroke for increase of load and earlier for a decrease of load, the pressure in the valve chest remaining practically constant. Provision is made for varying the normal running speed of the engine while the engine is actually in operation, and this is done by means of the hand wheel shown on the throttle valve spindle.

The remaining details only call for brief notice. It will be observed that the cylinder

is mounted on a substantial cast-iron standard placed at the back of the engine and carrying the crosshead guide. This standard is securely bolted to the base plate, which is also of cast iron. Supporting the front of the cylinder is a polished steel pillar also secured to the bed plate, thus leaving the front of the engine open for lubrication and inspection. An oil reservoir is mounted on the top of the engine standard at the front, and pipes are run to all the working parts, the supply of oil to each pipe being regulated by a separate cock. The base plate is so shaped that it forms a crank chamber into which the oil drains after use, whence it can be drawn off when necessary through a small cock for filtering. The lubrication of the main crank shaft bearings is from drip feed oil cups. Finally, it is to be noted that a heavy flywheel is provided to ensure

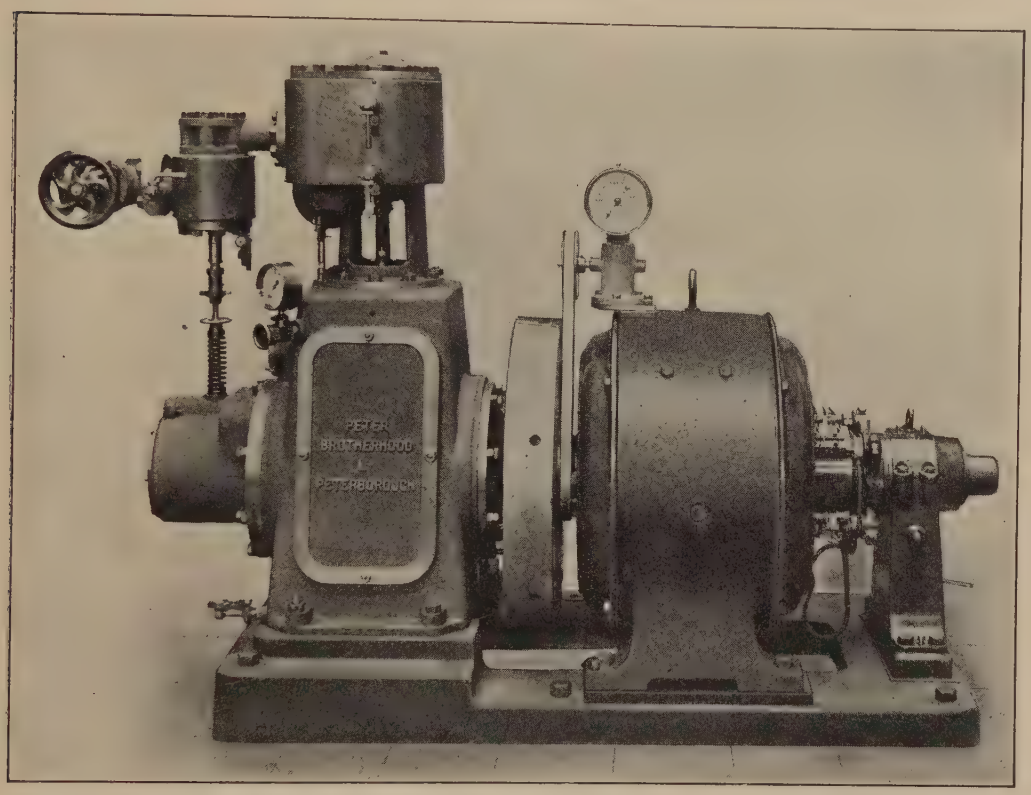


FIG. 15.—High-speed Enclosed-type Vertical Engine by Messrs. Peter Brotherhood, Ltd., Peterborough, driving a 20 kw. Dynamo at 650 revs. per min.

steady running and to act as a belt pulley if desired. Engines of this type are particularly reliable, and if sufficient lubricating oil is provided, they will run continuously for long periods without attention.

Another form of reciprocating steam engine which in its essentials is not greatly different from the above is the enclosed-type high-speed vertical engine. For driving electric generators, rotary pumps, and fans, it is essential for reasons of economy that the speed of rotation should be as high as possible, and while the open type of engine can be designed to give very good service at these speeds, the problem of lubrication becomes increasingly difficult. To overcome this difficulty the crank shaft and the connecting rod together with the crosshead and its guides are very often totally enclosed and a system of forced lubrication adopted. A small pump, usually driven by an eccentric on the main crank shaft, is installed, and this draws oil from a "sump" in

the crank case, forcing it through all the bearings. The excess oil drains from the sides of the bearings into the crank case, being collected in the sump, from whence it is drawn by the pump through a filter and used over and over again. The open type of engine would not be suitable for forced lubrication, as the excess of oil which is forced through the bearings would be flung about the engine room ; but when the working parts are totally enclosed, the splashing of the oil tends to add to the efficiency of the lubrication.

As an example of a single-cylinder, high-speed, enclosed vertical engine we may consider Fig. 15, which is a reproduction of a photograph of an engine of this type designed and constructed by Messrs. Peter Brotherhood, Ltd., of Peterborough. These engines are available in powers ranging from 2½ b.h.p. up to 120 b.h.p., the speeds ranging from 750 revs. per min. down to 500 revs. per min., the smaller units having the higher speeds. The unit shown in the illustration is direct coupled to a direct current generator having a capacity of 20 kw., the power of the engine when steam at 120 lb. per sq. in. is supplied at the stop valve being about 28 b.h.p. and the speed of rotation 650 revs. per min. The principal features of the engine do not differ materially from those of the open type already described, the cylinder and valve chest forming a single casting which is bolted on to the main casing, this in its turn being bolted on to the base plate. The casing is provided with a large inspection door to give access to the various parts for cleaning and adjustment, while the main crank shaft bearings are so designed that they can be removed after detaching the circular plates on the sides of the casing, without removing the crank shaft. When required, however, the crank shaft is removed through these side doors.

FIG. 16.—Eccentric Strap and Rod for Peter Brotherhood High-speed Enclosed Engine.

Distribution of the steam is effected by means of a slide valve of the piston type driven by means of an eccentric in the usual way, the valve spindle working through a guide bored in the casing. The eccentric strap, which together with its rod is shown in Fig. 16, is made of bronze, the lower half being drilled as shown to take the rod which operates the plunger of the lubricating oil pump. The eccentric rod is made of steel, the top end being fitted with a phosphor-bronze bush to take the pin which connects it to the valve rod. The piston in all but the smallest units is made of forged steel and is of dished form to facilitate the drainage of any water which may condense upon it. The piston rod, which is of steel, carries the two crosshead guide slippers and the gudgeon pin bearing at its lower end. The crosshead guides, which are of circular form, are bored through the top of the casing, being thus accurately in line with the centre of the cylinder. The piston rod with the crosshead is shown in Fig 17. A short connecting rod of the marine type is employed, the main features of which can be easily understood from Fig. 18, which shows the piston rod and the connecting rod assembled together. A pump is provided which delivers the lubricating oil at a pressure of from 15 to 25 lb. per sq. in. to the various bearings, a gauge being provided (seen on the left in Fig. 15) to indicate the pressure of the oil on the delivery side of the pump. The

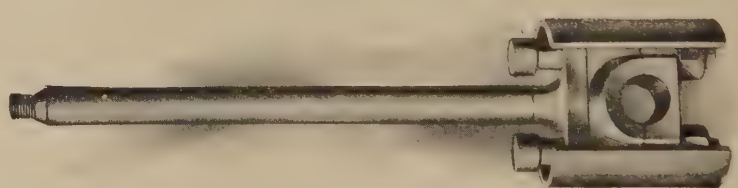


FIG. 17.—Piston Rod and Crosshead for Peter Brotherhood High-speed Enclosed Engine.

pump, which is of the oscillating type thus eliminating all valves, is fitted with a strainer on the suction side, and delivers oil to all the main bearing surfaces, while in addition to the positive supply from the pump the splashing of oil in the casing ensures perfect lubrication of all parts. It may be noted that the cylinder is mounted at a distance

Fig. 18.—Piston Rod and Connecting Rod for Peter Brotherhood High-speed Enclosed Engine.

above the casing which is greater than the stroke of the piston, so that no oil is brought into the cylinders by the piston rods, as this might lead to trouble in the condensers.

The governor is of the centrifugal type mounted on the end of the crank shaft and works in an oil-tight casing, the moving parts being directly lubricated. It actuates a double-ported regulating valve in the manner already described. A sectional view of this valve is shown in Fig. 19. It is claimed by the makers that the speed can be controlled to within $2\frac{1}{2}$ per cent. or even less under sudden variations of the load from zero to the maximum and *vice versa*. In the case of the larger engines, an additional safeguard is provided in the form of a trip governor which entirely cuts off the supply of steam in the event of the speed rising to a definite figure above the normal. When the engine is employed for purposes other than driving an electrical generator, e.g. for driving fans or centrifugal pumps, it is necessary to be able to run the engine at various speeds, and the centrifugal governor is therefore not fitted; it is, however, still necessary to fit the trip governor in order to guard against the possibility of the engine getting out of control.

Fig. 19.—Sectional View of Double-ported Governor Valve for Peter Brotherhood High-speed Engine.

So successful has this type of engine been in practice that it has been found advantageous to construct units which, while not differing greatly as regards output from those just described, are nevertheless able to work with greater economy than is possible with a single-cylinder unit. It has already been pointed out that considerable saving from the point of view of steam consumption, which is, of course, synonymous with fuel consumption, is obtained by allowing the steam to expand to the fullest extent, and that it is preferable that this expansion should occur in two or more cylinders, thus yielding a compound engine. Such engines are necessarily more complicated than the single-cylinder engine, and the designer is compelled to compromise between economy in cost of construction and economy in cost of running. For the smallest powers the balance is generally on the side of the former, and under these circumstances the single-cylinder engine is largely used for generating small powers. By the adoption of the tandem system, that is by placing two cylinders in line, it is possible to have a compound engine without the complication of a multiplicity of cranks, connecting rods, etc., and Messrs. Peter Brotherhood have adopted this idea

with conspicuous success. A good idea of the arrangement of these tandem compound engines can be gathered from Fig. 20, which shows a unit of about 55 b.h.p. direct coupled to an electric generator of 40 kw. capacity.

From a study of this illustration it will be seen that the fundamental difference between this type of engine and the single-cylinder engine is that the former has an extra cylinder mounted above the first cylinder. The high-pressure cylinder is uppermost and the steam enters this cylinder via the governor valve in the usual way. After expanding and doing work in this cylinder, the steam passes back to the lower portion of the valve chest, where it is distributed by a second slide valve to the lower cylinder. In this cylinder, which is of course of larger diameter than the first, owing to the lower pressure and therefore the larger volume that has to be dealt with, the steam

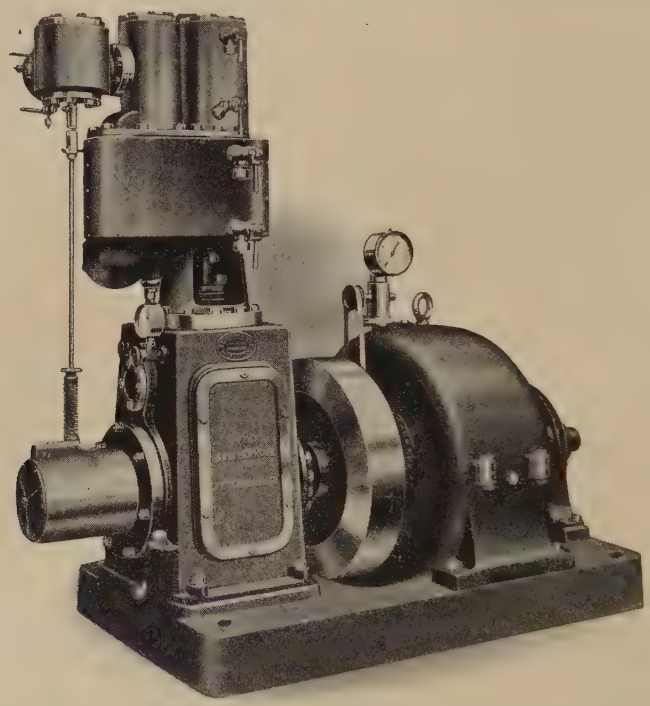


FIG. 20.—Peter Brotherhood Tandem Compound Engine of 55 b.h.p., direct coupled to a 40 kw. Electric Generator.

expands, doing more work until it finally passes to the exhaust pipe, the flange of which can be seen below the low-pressure cylinder. Fig. 21 shows a sectional side elevation and a sectional end elevation of this engine, from which it will be seen that the high-pressure and the low-pressure pistons are mounted on the same piston rod, the lower end of which carries the crosshead, this being of the same form as for the single-cylinder unit. The two valves are mounted on one spindle and operated by a single eccentric, the general features of the engine being very similar to those already described. The details of the forced lubrication oil system will be easily understood from a study of the sectional side elevation. The pump is seen immediately beneath the eccentric, and the delivery pipes to the main bearings are clearly shown. The connection to the crosshead guides is, however, not seen, although the distributing system from the gudgeon pin to the main crank pin bearing is visible. Attention can also be drawn to the governor gear, the details of which are visible on the left of the side elevation.

Simple as the tandem compound engine undoubtedly is, it suffers from the disadvantage inseparable from all single-crank engines, namely, that it cannot be perfectly balanced. Heavy flywheels can be provided and balance weights may be attached to the crank webs, but there are always certain out-of-balance forces acting upon the engine when it is in motion, and these set up vibrations in the engine which become more and more pronounced as the speed increases. In a well-designed and carefully-made engine these vibrations may be very small indeed, but nevertheless they are always present. In order to overcome this disadvantage as far as possible and particularly for large powers or in circumstances where two or more engines driving electrical generators which feed into the same system are employed, it is found advisable to employ a two-crank

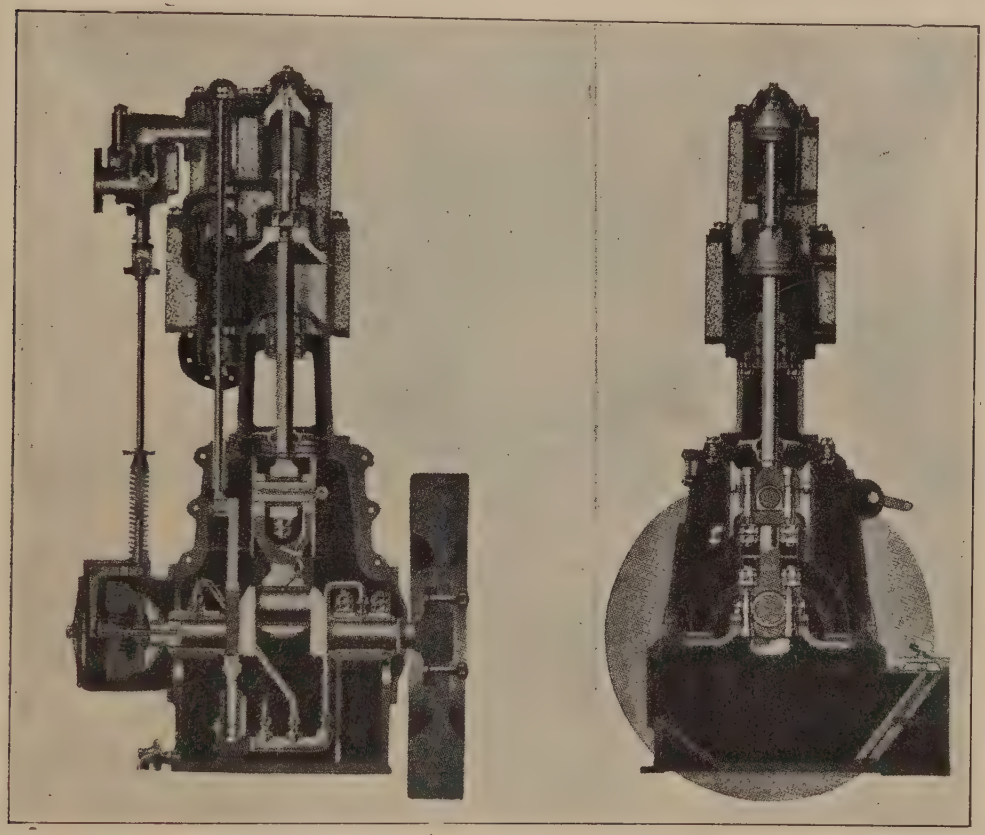


FIG. 21.—Sectional Views of Peter Brotherhood Tandem Compound Engine.

engine, the high-pressure and the low-pressure cylinders being mounted side by side. The arrangement then becomes as shown in Fig. 22, which is reproduced from a photograph of a high-speed, vertical, two-crank compound engine running at 400 revs. per min., designed and constructed by Messrs. Peter Brotherhood, and connected to an electric generator of 200 kw. capacity. The crank shaft of this engine is seen in Fig. 23, from which it will be noticed that the cranks are set at 180° to each other, and not at 90°, as might be expected if the question of a uniform turning moment on the shaft or facility of starting up the engine is considered. The cranks are set at 180° to each other in order to ensure that the steam from the high-pressure cylinder shall be exhausted just at the moment it is required to enter the low-pressure cylinder, and this occurs for the downstroke of the l.p. piston when the h.p. piston is at the bottom of the cylinder and

vice versa. The other details of these engines are substantially similar to those already described. A point of particular interest about the unit shown in Fig. 22 lies in the fact that it was designed and constructed by Messrs. Peter Brotherhood to the order of the British Admiralty, and is installed in H.M.S. *Hood*, the largest and most powerful warship ever built for the British Navy. The electric generator is of novel form, being

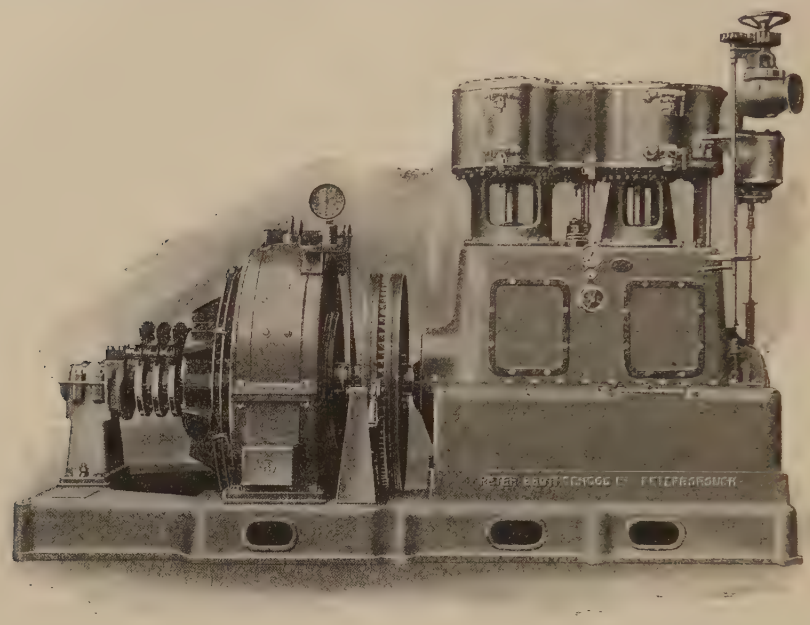


Fig. 22.—Peter Brotherhood High-speed Compound Engine driving a 200 kw. Generator on H.M.S. *Hood*.

wound to give direct current for lighting and power purposes and also to give three-phase current for wireless telegraphy purposes. It will be noted that the engine and the generator are mounted upon a common base plate of substantial proportions. This ensures correct alignment of the engine with the generator and is the usual practice with marine sets. With land sets the engine and the generator are often separately secured to the foundations, as there is little or no risk of any relative movement. It may be noted

Fig. 23.—Crank Shaft for Peter Brotherhood Two-cylinder Compound Engine.

that Messrs. Peter Brotherhood have supplied many hundreds of these high-speed vertical engines of the different types described above to the various Government departments, chiefly to the Admiralty, for driving electric generators and forced-draught fans.

From the foregoing it might be imagined that the high-speed vertical engine is only used for the development of low or comparatively low powers. Such, however, is far

from being the case. It can confidently be affirmed that but for the development of the steam turbine, this type of engine would have been almost universally employed for the driving of electrical generators, and many such units, some of them developing over 1,000 horse power, were actually built in the opening years of the present century– that is, before the reliability of the steam turbine had been fully established. These engines were generally of the triple-expansion, high-speed, vertical type, having three cranks, so as to obtain as perfect a balance as possible ; while in order to obtain the requisite cylinder volume the tandem principle was also employed in conjunction with a multiplicity of cranks, a high-pressure, intermediate-pressure, and low-pressure cylinder being superimposed one over the other, thus giving as many as nine cylinders per engine. The development of the steam turbine has rendered obsolete these high-powered reciprocating engines, and it is now possible to obtain single turbine units developing powers far in excess of anything ever achieved by the reciprocator. For intermediate powers—say up to about 1,000 horse power—the latter type of engine is still extensively used at the present day and many different designs are still being built ; it will be of interest therefore to consider one or two examples of reciprocating steam engines having powers greater than those of the units so far described. We have frequently laid stress upon the fact that for the generation of electrical energy it is economical to employ units having a high speed of rotation, and hence we find that when used for driving electrical generators, reciprocating engines having a high speed of rotation are generally employed. The type of engine usually adopted may be regarded as a direct development of the low-powered, high-speed, vertical engines to which attention has already been drawn.

The firm of Belliss & Morcom, Ltd., of Birmingham, have for many years occupied a very prominent position as manufacturers of high-speed reciprocating steam engines, in fact they were the pioneers of the idea of forced lubrication involving the enclosing of the crosshead, connecting rod, and crank shaft of the engine, their patent having been taken out in the year 1890. In that year they built an engine of 20 horse power embodying their ideas which had a speed of 625 revs. per min. ; and as indicating the length of life which perfect lubrication gives, it is interesting to note that this engine was in constant use at the works in Birmingham until the year 1919. It is now being preserved because of its historic interest From the original engine Messrs. Belliss & Morcom have evolved a number of different designs, some of them capable of developing very high powers for this type of unit. Considering first of all a reciprocating engine of intermediate power, their V-type engine—so called on account of the arrangement of the slide valves—is worthy of a detailed description. This engine is of the two-crank compound type, made in sizes varying from 120 b.h.p. up to 570 b.h.p.—that is, for driving generators of capacity varying from 80 kw. to 400 kw., the speed of rotation of the smaller units being 575 revs. per min. and that of the larger units 375 revs. per min. The general appearance of these engines can be gathered from a glance at Fig. 24, which shows one of these engines direct coupled to an electric generator.

In order to fully understand the details of the design of these engines, reference must be made to Fig. 25, in which a sectional front elevation and a sectional end elevation are shown. It will be seen from the front elevation that the two cylinders are arranged with their centre lines vertical with the cranks at 180° to each other. The slide valves are set between the cylinders, there being two eccentrics driving the two valves, which are inclined to each other as shown in the end elevation, so that the valve rods form a letter V, whence the term " V "-type as applied by the makers to these engines. The valves work in separate chambers, this being considered an advantage as compared with a not unimportant type of engine in which distribution of the steam is effected by two valves mounted upon a single spindle and working in a common chamber. The slide valves are of the piston type of the form shown, and the method whereby they dis-

tribute the steam from the high-pressure cylinder to the low-pressure cylinder is clearly indicated by the arrows. The pistons are of the cone type and are fitted with metallic packing to prevent the leakage of steam from one side of the piston to the other, while the lower cylinder covers are provided with specially designed stuffing boxes also fitted with metallic packing, as are the valve chest covers. The distance between the top of the casing and the underside of the cylinders is made greater than the travel of the piston, so as to prevent any possibility of oil being drawn up into the cylinders, while in order to prevent the leakage of oil past the casing top at the points where the piston rods and valve rods emerge, scraper rings are fitted as shown.

The question of lubricating the various bearings and guides has received very careful attention, forced lubrication being employed in accordance with the principles originated

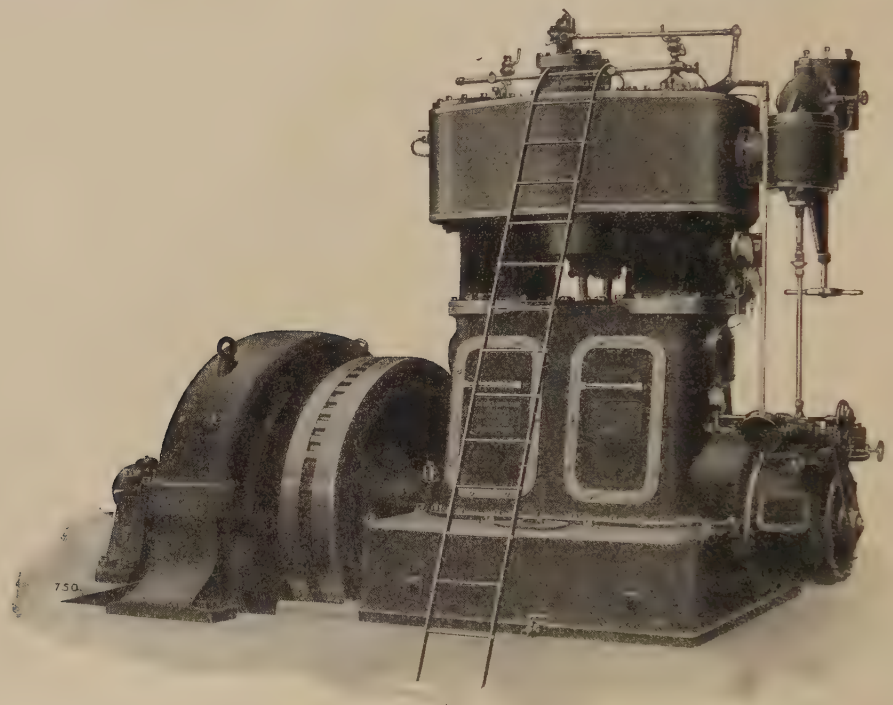


FIG. 24.—High-speed Vertical Compound "V" type Engine by Messrs. Belliss & Morcom, Ltd., direct coupled to an Electric Generator.

by the makers. A valveless oil pump driven from one of the eccentrics is employed and the oil is distributed from the delivery of the pump to all the working parts of the engine through a specially arranged system of pipes and channels. The direction of the lubricating oil is indicated by a series of arrows in both views of Fig. 25. It will be seen that direct delivery is made to the crank shaft bearings, of which there are four, and that from the two intermediate bearings through a hole drilled from the crank shaft through the web to the centre of the crank pin, oil is fed to the connecting rod bottom end bearings. A hole is also drilled through the centre of the connecting rod diverging into two at the forked end of the rod, and oil from the bottom end bearing passes up this hole to the two small gudgeon pin bearings. In this way oil under pressure is led to all the bearings on the connecting rod. A somewhat similar arrangement is adopted for lubricating the bearing surfaces on the eccentric and its rod, a hole being drilled through the crank shaft to the surface of the eccentric sheave, so that oil from the intermediate main

bearings is fed to the eccentric straps. Instead, however, of having holes through the centre of the eccentric rods, these being of small diameter in comparison with the connecting rod, an oil pipe is run from the eccentric strap up to the brushes at the forked

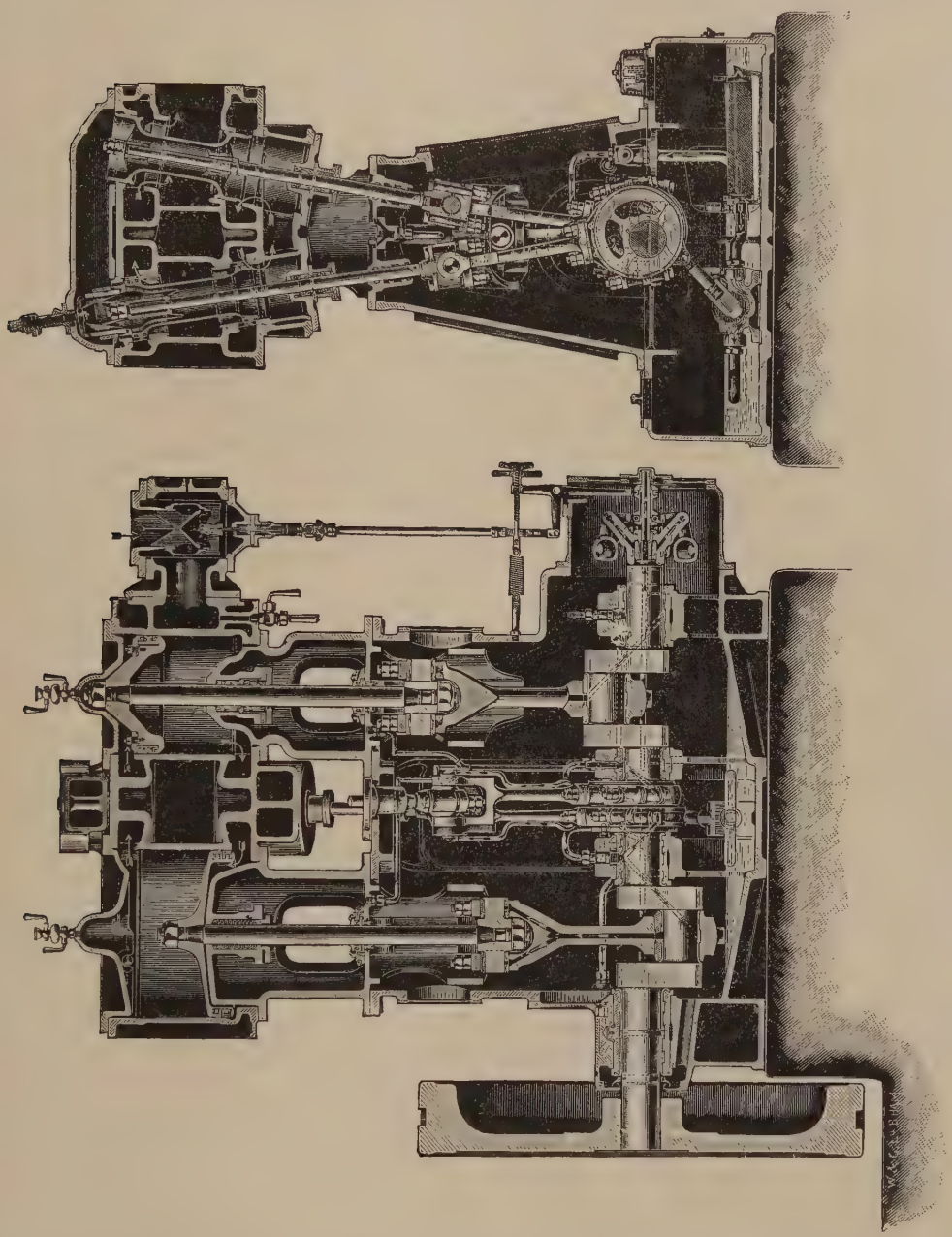


FIG. 25.—Sectional Views of "Belliss" V-type High-speed Vertical Compound Engine.

end of the eccentric rods. It should be noted that the main crosshead guides and the valve rod guides form part of the casting which constitutes the support and distance piece for the cylinders. The top portion of this casting also forms the cylinder covers, and the guides can thus be bored out concentric with the cylinder covers and the stuffing boxes, thereby ensuring perfect alignment of the crosshead and the cylinder. It will

be seen that an arrangement is fitted on the suction side of the oil pump which automatically separates from the oil any water which may find its way into the casing from the glands, the water going to waste and the oil passing to the filter.

The engine is fitted with a centrifugal governor operated on a spring drive principle which has been patented by Messrs. Belliss & Morcom, and by means of which it is claimed that wear-and-tear is reduced to a minimum. As will be seen from the sectional front elevation in Fig. 25, the governor operates a throttle valve which gives under ordinary conditions a variation of speed which is guaranteed not to exceed 3 per cent. between full load and no load. By means of the small hand-wheel shown just above the governor, small variations in the speed can be made when the engine is running. When the highest economy of steam consumption is required at all loads, even to a possible 25 per cent. overload, or when the engine may be called upon to run for extended periods at light loads, a patent form of variable-expansion governor is fitted. This device is not shown in Fig. 25, but the rods leading from the governor at the end of the crank shaft to the high-pressure valves can be seen in Fig. 24.

We have already seen that a very large increase occurs in the amount of work done in the cylinder of an engine per lb. of steam as the cut-off is made earlier and earlier, although at the same time the work done during each stroke of the piston becomes less and less. If, however, owing to a reduction in the load on the engine, it is necessary that the work developed by the engine should also be reduced, it follows that by making the point at which the supply of steam to the cylinder is cut off earlier—that is, by allowing a smaller quantity of steam to enter the cylinder, such steam expanding to a greater extent than at full load—a very economical method of operating the engine will result. If, moreover, this change in the point of cut-off can be effected automatically with variation of the load, an efficient form of governor will be obtained. This is what is done in the " variable-expansion " type of governor. Instead of reducing the pressure of the steam at the main stop valve, as is done in the case of the throttle governor, when the load increases the point of cut-off is automatically changed, so that it occurs later in the stroke, while for a decrease in the load an earlier cut-off is effected, the pressure of the steam at the stop valve of the engine remaining constant. It can be demonstrated that the variable-expansion governor is a more economical type than the throttle governor—that is, the work done per lb. of steam is greater for different loads on the engine with the former type of governor than with the latter. On the other hand, the mechanism of the throttle governor is simpler than that of the variable-expansion type, particularly when wide ranges of load have to be dealt with. As in many other things, a compromise between the two types yields the most satisfactory result from all points of view, and hence we find that on the Belliss engines, except for the smallest powers, a combined throttle and variable-expansion governor is fitted, governing being effected by varying the cut-off for the higher loads, while at lower loads the speed of the engine is controlled by the throttle governor.

Variation of the point of cut-off is effected by rotating the slide valve, the steam ports being inclined to the plane perpendicular to the axis of the valve. The cut-off can be varied from one-half to eight-tenths of the stroke, which means, of course, a wide variation in the power developed per working stroke of the piston. For fluctuations of the load up to about three-quarters full load, governing is entirely by variation of the opening of the throttle valve, the cut-off remaining fixed at about one-half stroke until the full open position of the throttle is reached. On further loading up the engine, the throttle remains in the full-open position, but the point of cut-off becomes later and later in the stroke, until at full load or at the desired overload a cut-off at eight-tenths of the stroke occurs. The throttle valve also continues to open, but in such a way that it has no effect upon the pressure of the steam passing to the engine. In the event of the

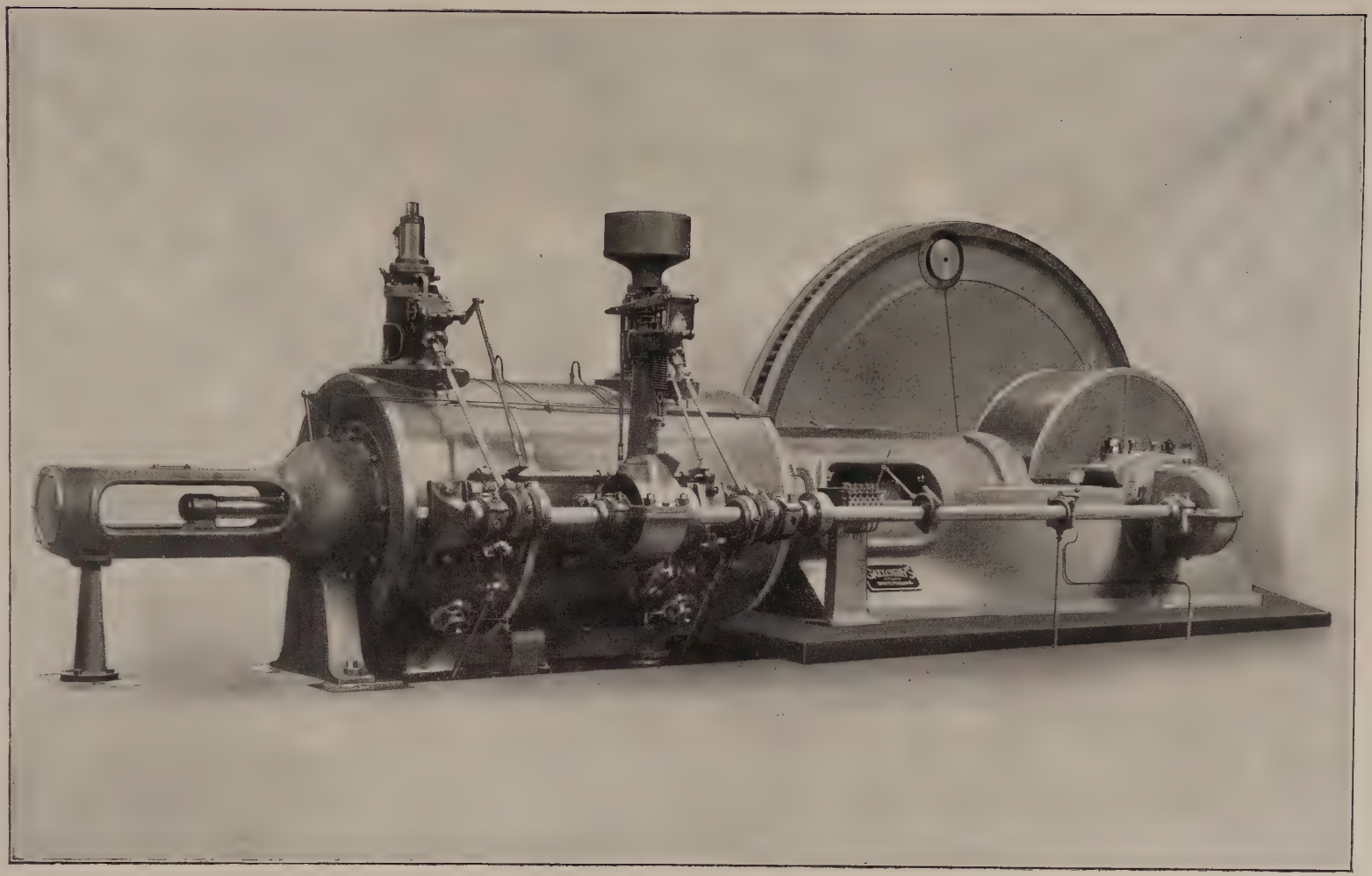


PLATE I.—SINGLE-CYLINDER HORIZONTAL DROP-VALVE ENGINE BY MESSRS. GALLOWAYS, LTD., MANCHESTER.

load being suddenly thrown off the engine, the throttle will instantly close, but the expansion gear which operates through a spring will only gradually rotate the valve, thus preventing any damage due to too sudden movement of the various parts of the mechanism. It should be noted that if desired the variable-expansion gear can be disconnected while the engine is running, the engine being governed by the throttle only.

A very good idea of the value of the variable-expansion governor gear, in so far as it affects steam consumption, can be gathered from a study of Fig. 26, which shows two typical steam consumption curves, not indeed for the "V" type of engine, but for a "Belliss" triple-expansion engine of 1,080 brake horse power. This engine was tested first with the variable-expansion gear in operation and then with the expansion gear disconnected, the steam pressure in each case being 190 lb. per sq. in. at the stop valve, superheated 150° F., and exhausting at a vacuum of 27 in. of mercury. It will be noticed that at all loads from one-half full load up to a 20 per cent. overload the steam consumption with the variable-expansion gear in operation is superior to that when governing is by throttle only, and that while at full load the difference is not very great, it becomes more and more pronounced in favour of the variable-expansion gear both above and below full load. It may be noted in passing that at full load the engine shows the very satisfactory performance of 11·75 lb. of steam per b.h.p. per hour, a figure which compares favourably with that obtained from many turbines of the same output. For a two-cylinder compound engine it is unlikely that so low a steam consumption figure would be obtained, but at least as far as the system of governing is concerned the relative effect of the two systems would be very much the same.

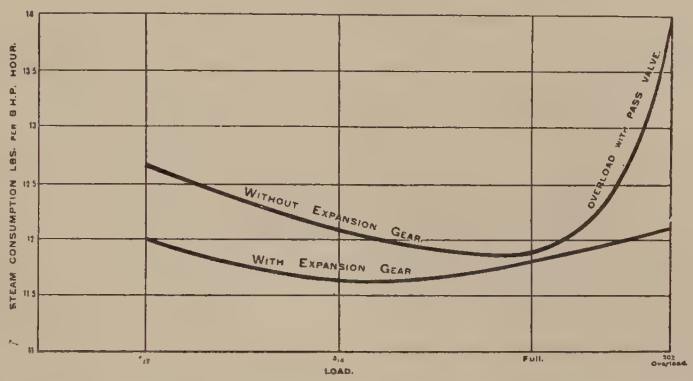


FIG. 26.—Steam Consumption Curves for "Belliss" Triple-expansion Engine of 1,080 b.h.p. with and without Expansion Gear.

Before passing on to a detailed description of the triple-expansion engines referred to in the previous paragraph, it will be of interest to draw attention to one or two typical installations in which the V type of engine built by Messrs. Belliss & Morcom, Ltd., is to be found. It may be mentioned in passing that since the makers introduced this type of engine in the year 1908, they have sold 1,164 units, aggregating 362,000 horse power. Fig. 27 shows what may be considered as a typical works power station, such as will be found in many up-to-date industrial establishments. The works in question is a large chemical manufactory, and the plant comprises two V-type engines each of 570 b.h.p., having a speed of rotation of 343 revs. per. min. and supplied with steam at a pressure of 150 lb. per sq. in. The engines are run non-condensing, the pressure of the exhaust being 5 lb. per sq. in. above atmospheric pressure. The exhaust steam is utilised for various purposes in the works, and it is found to be a highly efficient procedure to generate steam at high pressure, employ this steam for driving electrical generators, and then use the exhaust at approximately atmospheric pressure for the various chemical processes in the works, much more efficient, indeed, than to generate steam at atmospheric pressure and to obtain the electric power by means of engines and dynamos of the usual type

It has already been demonstrated that the steam turbine is far more efficient at

extremely low pressures than is the steam reciprocating engine, and hence in circumstances where a certain amount of low-pressure steam is available turbines can be used in conjunction with reciprocating engines to very great advantage. Such an installation is shown in Figs. 28 and 29, which are reproductions of photographs of the power house of a firm of manufacturers of railway rolling stock. There are two " Belliss " V-type engines driving electrical generators of 400 kw. capacity each, the engines receiving steam at a pressure of 165 lb. per sq. in. and exhausting it at a pressure of 16 lb. per sq. in. absolute (i.e. about 1·3 lb. per sq. in. above atmospheric pressure). To each of these engines there is a mixed-pressure turbine,[1] as shown in the illustration, into which the exhaust from the reciprocating engine is passed, while as required these turbines are supplied with boiler steam. In these turbines, which together with their condensing plants are also of Belliss & Morcom's manufacture, the steam expands to

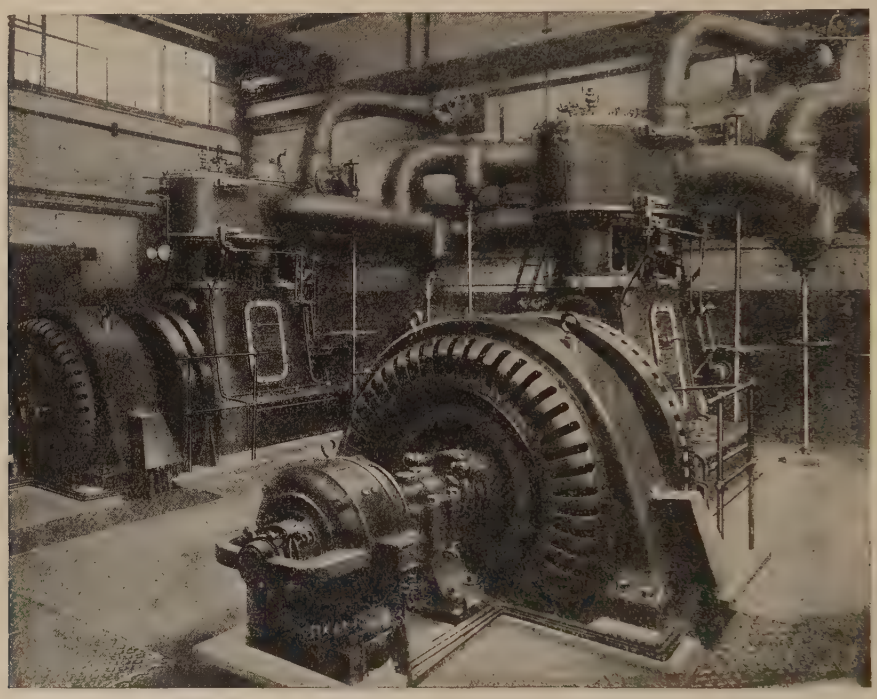


Fig. 27.—Two " Belliss " 570 b.h.p. V-type Engines installed at a Chemical Works. The exhaust from the engines is used for various purposes in the works.

the lowest possible pressure that can economically be obtained in the condenser. A combination of this sort presents many points of advantage, due largely to its flexibility. Thus, the engine can be used alone, either condensing or exhausting to the atmosphere, while the turbine can also be run independently with live steam if desired. The turbines can also be run on the exhaust from the various steam hammers, etc., which are usually found in a works of this nature. It may be noted that the addition of a similar exhaust turbine to the triple-expansion engines at the power house at Messrs. Belliss & Morcom's own works, the turbine also utilising the exhaust from the steam hammers, has resulted in a reduction of 20 per cent. in the coal consumption. This gives a very good idea of the savings which can be effected by the use of exhaust steam turbines, particularly when steam is employed which would otherwise be condensed directly or possibly wasted.

[1] For a full description of the mixed-pressure type of turbine see Volume I, Chapter VI.

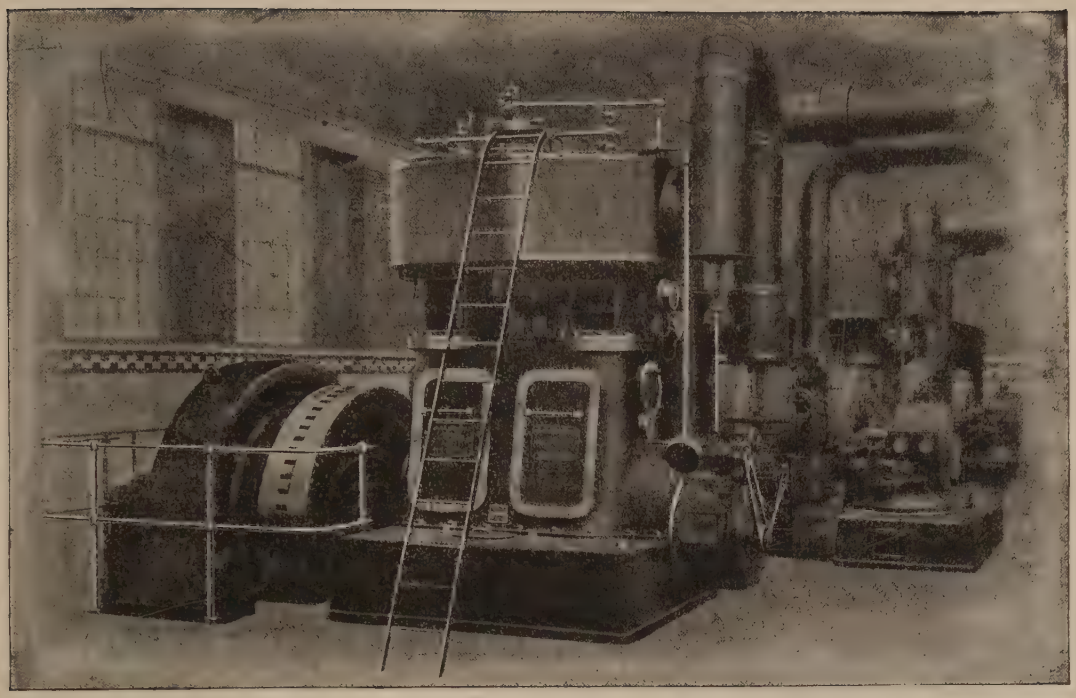


FIG. 28.—Two " Belliss " V-type Engines driving 400 kw. Generators, each exhausting into a Mixed-pressure Turbine.

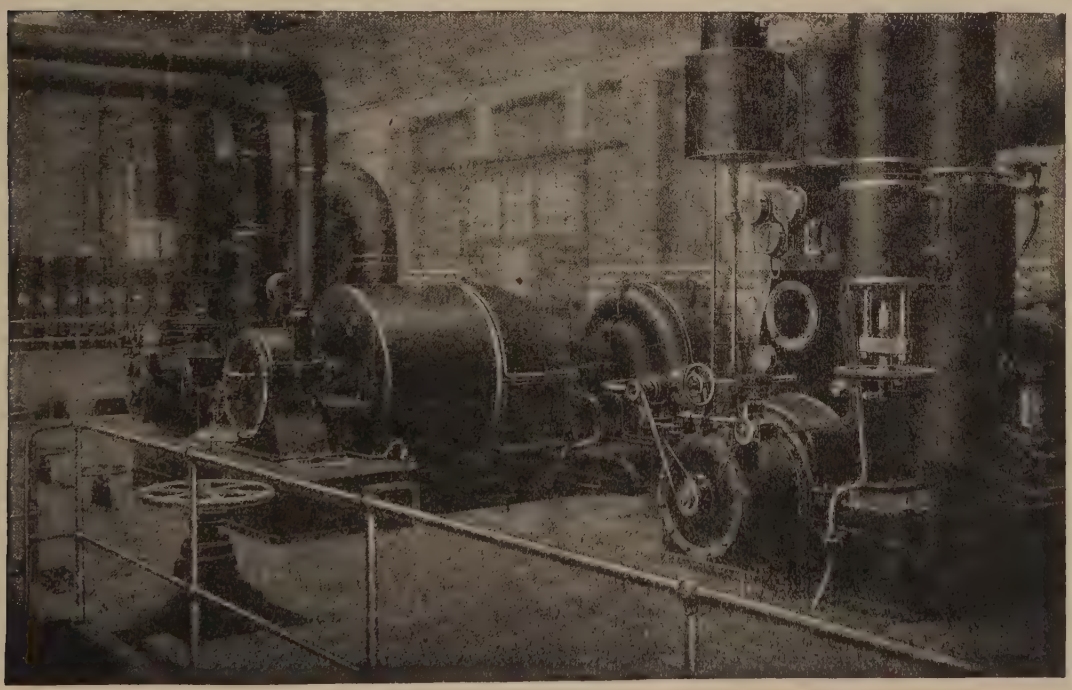


FIG. 29.—Another View of the " Belliss " Engines and Mixed-pressure Turbines, all driving Electric Generators.

We come now to a consideration of reciprocating engines of the largest powers for which this type can be economically employed. As might be inferred from what has already been said, the triple-expansion principle is generally adopted, the steam expanding in a "high-pressure cylinder," from which it passes to an "intermediate-pressure cylinder," where it is expanded further and from which it passes to a "low-pressure cylinder," expanding still further before it is finally exhausted into the condenser. As the steam passes through the engine its volume increases, and hence the diameter of the cylinders must be progressively increased, as indeed must all the steam passages, valve chests, ports, etc., throughout the engine. Where high speeds are desirable, as for the driving of electrical generators, the vertical self-lubricating type of engine is generally employed, and as far as the main features of the design are concerned the engine can be considered as similar to the compound engines previously described with the addition of another cylinder. There are now three cranks, which, in order to provide a uniform turning moment on the shaft while keeping the volume of the steam spaces

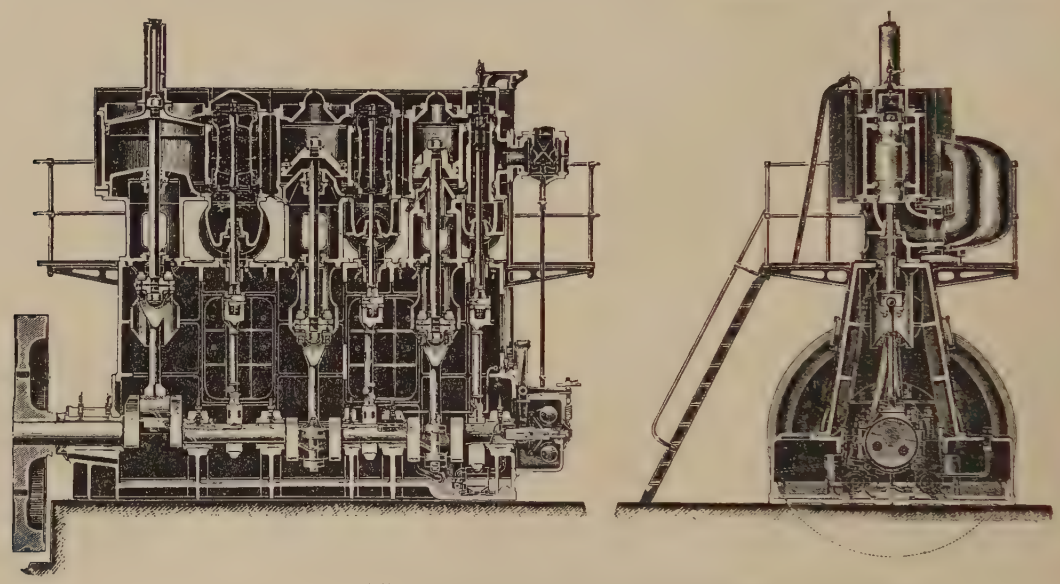


FIG. 30.—Sectional Views of "Belliss" Triple-expansion High-speed Vertical Engine.

or "receivers," as they are termed, between each successive pair of cylinders as small as possible, are arranged at 120° to each other.

As a typical example of a high-speed, triple-expansion engine, we can consider the unit shown in Fig. 30, which is a section of the "Belliss" patent self-lubricating type of triple-expansion engine. Engines of this type are built in sizes varying from 260 maximum b.h.p. up to 2,500 maximum b.h.p., these powers being equivalent to an electrical output of 180 kw. up to 1,750 kw. The smallest units have a maximum speed of 550 revs. per min., while the largest units run at a speed of 200 revs. per min. The general appearance of these engines can be gathered from Fig. 31, which shows two of them direct coupled to alternating current generators. These engines have a speed of 250 revs. per min. and develop 1,300 to 1,625 b.h.p., the steam being supplied at 175 to 180 lb. per sq. in. pressure superheated to a temperature of 550° F. The size of these units can be gauged in comparison with the man standing on the platform by the low-pressure cylinder. Between the two units will be seen two turbo-generators, these being of the mixed-pressure type driving generators of 500 kw. capacity each, the turbines together with their condensing plant having also been supplied by Messrs. Belliss & Morcom.

Considering again the sectional view of the engine (Fig. 30), it will be noticed that the arrangement of the crossheads and connecting rods is substantially the same for each cylinder. The high-pressure piston has of necessity to be made stronger than the other two, since it is subjected to steam at a higher pressure. For the same reason the piston packing in the high-pressure cylinder is of special design, the rings being of the maker's patent self-adjusting type. The piston rod packing and gland may also be noted. As before, the cylinders are set at a distance in excess of the stroke of the piston above the casing in order to prevent the possibility of oil being carried into the cylinders, and special scraper rings are fitted where the piston rods pass into the casing for the double purpose of preventing oil from passing out of and water from entering into the casing. The crank shaft is supported on no less than six bearings, two between each crank and one at each end of the casing, that nearest the flywheel being of particularly

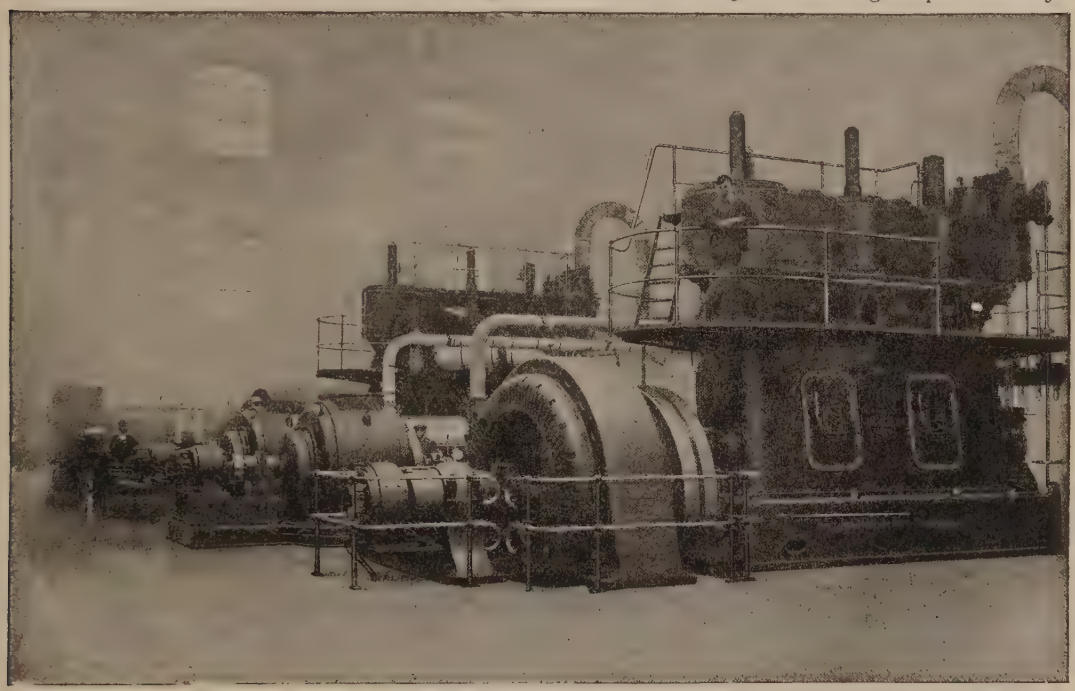


FIG. 31.—Two " Belliss " Triple-expansion Engines of 1,625 max. b.h.p., driving Alternating Current Generators at 250 revs. per min.

massive construction. All bearings are lubricated under pressure, two oil pumps being provided, both driven by the high-pressure eccentric, either of which is capable of maintaining the supply of oil to the various bearings. The oil after it has passed through the bearings falls into the crank chamber, from whence it drains into a sump immediately beneath the oil pumps. Strainers are arranged on either side of the pumps as shown, to which the oil pump suction is connected. The governor gear is of the combined throttle and variable-expansion type already described, the mechanism operating the variable-expansion gear on the high-pressure valve being clearly shown.

These engines have proved themselves, both on trial in the makers' works and in actual commercial operation, to be extremely efficient. Recognising the economies which result from using superheated steam, the engines are designed to run with a high degree of superheat, particular care being paid to the piston and piston rod packings. By the courtesy of Messrs. Belliss & Morcom, Ltd., we are enabled to publish the results of a series of tests obtained with a triple-expansion engine of the type just described. The

figures relate to an engine of 1,100 maximum b.h.p. running at 230 revs. per min., direct coupled to a generator of 750 kw. capacity, and tests were run at full load, three-quarter load, and half load. For the sake of comparison the results are given in British and also in metric units.

RESULTS OF TESTS OF "BELLISS" TRIPLE-EXPANSION SELF-LUBRICATING ENGINE USING SUPERHEATED STEAM

Load.	Full.	Three-quarter.	Half.
Steam Pressure, lb. per sq. in.	180	178	176
Kg. per sq. in.	12·69	12·52	12·40
Steam Temperature, °F.	600	605	608
°C.	316	318	320
Superheat, °F.	220	227	231
°C.	122	126	128
Vacuum at Engine, ins. of Hg.	25·7	26·0	26·0
mms. ,,	652	660	660
Speed of Engine, revs. per min.	230	230	230
Brake Horse Power	1,076	811	550
Dynamo Efficiency (assumed)	94%	93·5%	92%
Kilowatts	753	566	377
Consumption, lb. per b.h.p. hour	10·9	11·0	11·0
kg. ,, ,,	4·95	5·0	5·0
Consumption, lb. per kw. hour	15·55	15·7	16·0
kg. ,, ,,	7·07	7·14	7·27

The extreme economy of these units led to their extensive adoption for power station work, and prior to the advent of the steam turbine the high-speed triple-expansion provided the only equipment in a very large number of stations, the total power required being obtained by multiplying the number of sets installed. Of the many power stations equipped with "Belliss" engines the finest is probably the Summer Lane Electric Power Supply Station of the Birmingham Corporation Tramways. This station, a view of which

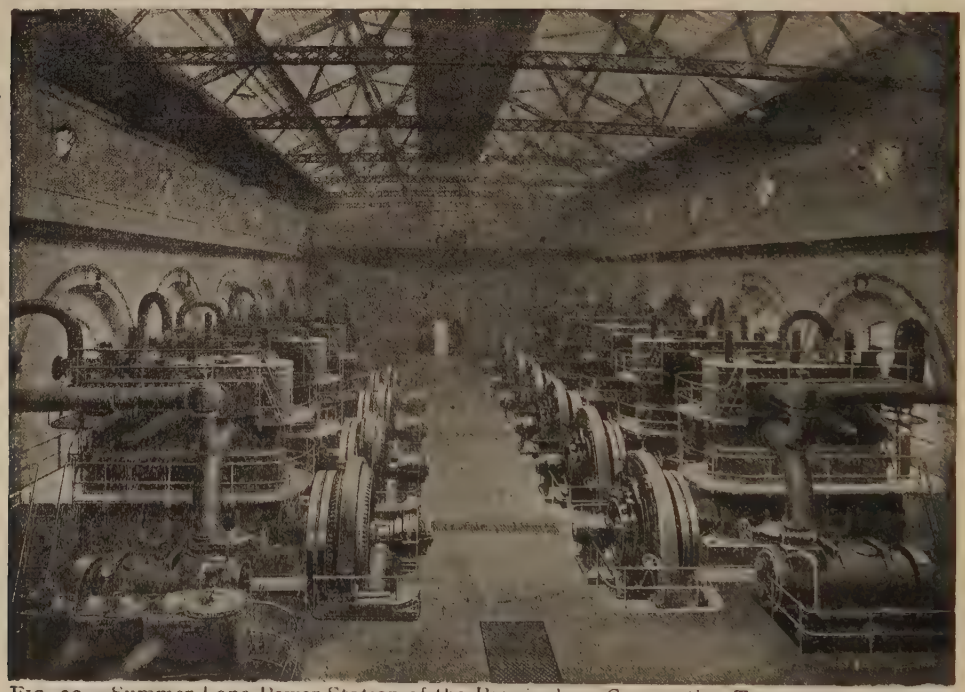


FIG. 32.—Summer Lane Power Station of the Birmingham Corporation Tramways, containing Twelve 2,650 max. b.h.p. Triple-expansion "Belliss" Engines.

is shown in Fig. 32, contains twelve engines, each of 2,120 to 2,650 b.h.p. having a speed of 160 to 166 revs. per min., steam being supplied at a pressure of 180 lb. per sq. in. Six of these sets are coupled to direct current generators of 1,500 kw. capacity, and six to alternating current generators also of 1,500 kw. In addition there are three smaller alternating current sets, each of 500 kw., and two direct current exciter sets, all of which are driven by " Belliss " engines. Each of the larger engines has its own surface condensing plant supplied by the engine makers, the pumps being alternately steam and motor driven. It may be noted that Messrs. Belliss & Morcom have also supplied six exhaust steam turbines to work in conjunction with a corresponding number of the larger engines. The addition of the turbine resulted in an increase in the total power from 1,500 kw. to 2,400 kw. per set, or 64 per cent., while at the same time there was a reduction in the steam consumption per unit generated of approximately 15 per cent. as compared with the consumption when the engine was running by itself and exhausting into the condenser as it was originally designed to do. When the turbines were installed, the cut-off in the different cylinders of the triple-expansion engines was adjusted so that while the power developed by the engine was the same as formerly, the exhaust from the low-pressure cylinder was at a pressure somewhat in excess of atmospheric.

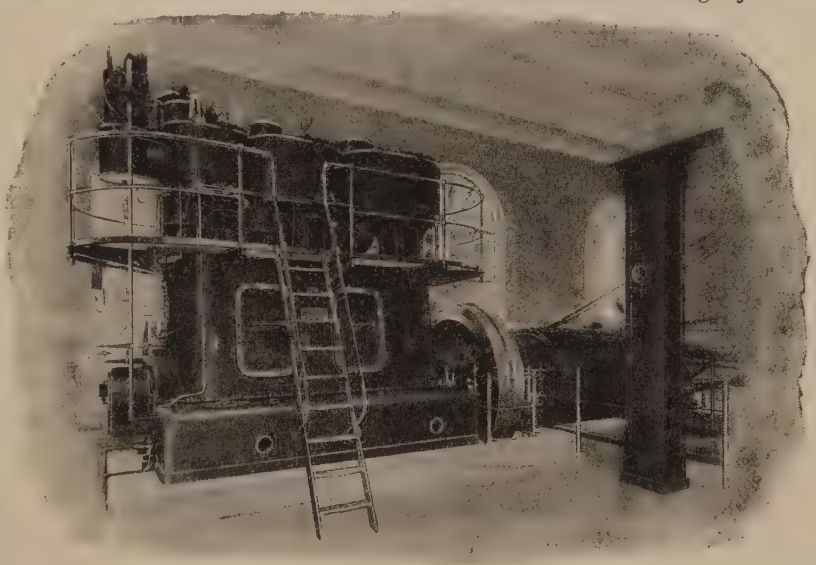


FIG. 33.—" Belliss " Triple-expansion Engine of 750 b.h.p. arranged for Rope Driving in a Cotton Mill.

Although the high-speed triple-expansion engine was originally designed for direct connection to electric generation, it has become recognised that it is particularly well adapted for driving the machinery in cotton mills, where ropes from the main pulley on the engine shaft drive the main pulleys on the shafting on the different floors of the mill. An example of a Belliss high-speed triple-expansion engine employed in this service is shown in Fig. 33. The engine is of 750 b.h.p. and runs at a speed of 300 revs. per min. A special rope pulley is mounted on an extension of the crank shaft, and the ropes to the pulleys on the shafts at each floor are driven by this pulley. These ropes can be clearly seen in the illustration. A high-speed engine is particularly suitable for use in conjunction with rope transmission, since high speed of rope is essential for an efficient drive, and with a slow-speed engine this can only be obtained by using pulleys of very large diameter. A further advantage of the high-speed triple-expansion engine lies in the fact that this type yields a very uniform speed of turning of the shaft, thereby increasing the output of the mill as well as the quality of the product. An uneven turning conduces to frequent breakages of the cotton with loss of time in stopping the individual machines for joining together again, and it has been found that the very steady turning has resulted in an increase of output of as much as 7 to 10 per cent. as compared with less efficient forms of drive.

CHAPTER III

THE HORIZONTAL RECIPROCATING STEAM ENGINE—THE DROP VALVE ENGINE—THE UNIFLOW ENGINE AND THE CORLISS ENGINE

IN the previous chapter we confined our attention to the vertical type of reciprocating engine, and we saw that this type had been developed for use on land where a high speed of rotation was required. For certain classes of work it frequently happens that a slow speed of rotation is very desirable, as, for example, in the case of driving steel rolling mills and the pit-head winding gear at a colliery. For these and for other purposes the slow-speed horizontal engine has been developed, and despite the competition of other forms of drive—notably the electric motor—this type of engine continues to be employed in very large numbers. The reciprocating steam engine possesses the very great advantage of being capable of running at very low speeds, i.e. when it is only just "turning over," to use an expression often heard in the engine room, without any serious falling off in the efficiency; further, it can be economically run at any desired speed between this low figure and the maximum designed speed. Another great advantage possessed by the reciprocating engine is the facility with which it can be made to run in either direction. It has already been noted that the steam turbine is essentially a high-speed engine and that it cannot be economically run at other than the designed speed, and, moreover, it cannot readily be reversed. The reciprocating steam engine thus possesses certain well-defined properties not possessed by the steam-turbine, and hence where slow and variable speeds are required, or where machinery is being driven which has to run in either direction, the former type of engine is more than able to hold its own.

The slow-speed engine must of necessity be a far heavier unit than the high-speed engine of the same power. This will be readily understood if we remember that power is measured by the rate of doing work, while work is the product of force and distance. Thus, measuring power in the practical unit, viz. horse power, we have

$$\text{HP} = \frac{\text{Work done per minute}}{33{,}300}$$

$$= \frac{\text{Force} \times \text{Distance moved through per minute}}{33{,}000}$$

In the high-speed engine the distance moved through by the force per minute is far greater than in the slow-speed engine, so that if the horse power is the same, the force exerted by the steam will be much smaller in the former case than in the latter, which means that the dimensions of the various parts and therefore the weight will be smaller. One result of this is to be found in the general practice of making slow-speed engines of the horizontal type and high-speed engines of the vertical type, for although the vertical engine occupies less floor space, if a very heavy engine were constructed with its cylinders vertical a very massive frame and base plate would be required. In the horizontal engine the weight is spread over a larger base, and hence a more rigid frame can be obtained with the least possible expenditure of material. For certain purposes, such

as pumping engines and in situations where space is limited, the vertical engine is to be preferred ; but generally speaking, where large powers are required and a slow speed of rotation is necessary, the horizontal reciprocating steam engine is employed.

Before describing in detail certain modern examples of engines of this type, we must first of all draw attention to one or two notable improvements which have been made in the design in recent years. Up to within comparatively recent times, the distribution of the steam in the cylinders of a reciprocating engine was effected by means of a slide valve, this being either of the flat or D type or of the piston type, similar to that employed in the engines described in the previous chapter. It may be noted in passing that the piston valve is exactly the same as the slide valve as far as the distribution of steam is concerned, the only difference being, as shown in Fig. 34, that in the former the working face of the valve forms a cylinder, while in the latter it forms a plane. The piston valve is preferred to the flat slide valve because it is balanced as regards the forces due to steam pressure and because it is easier to machine and fit accurately, a circular surface being always preferable to two flat parallel surfaces. On the other hand, the piston valve occupies far more space than does the flat valve, and as a result

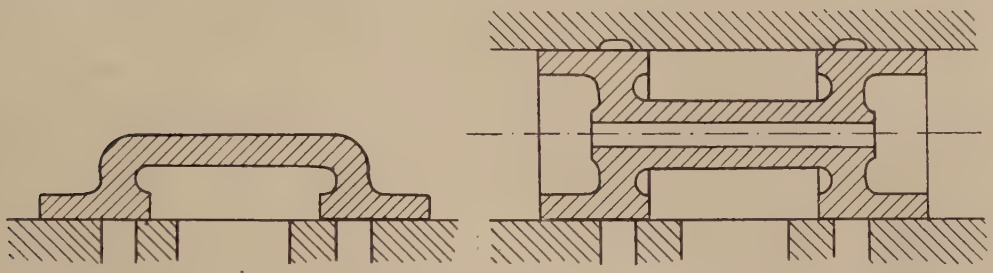


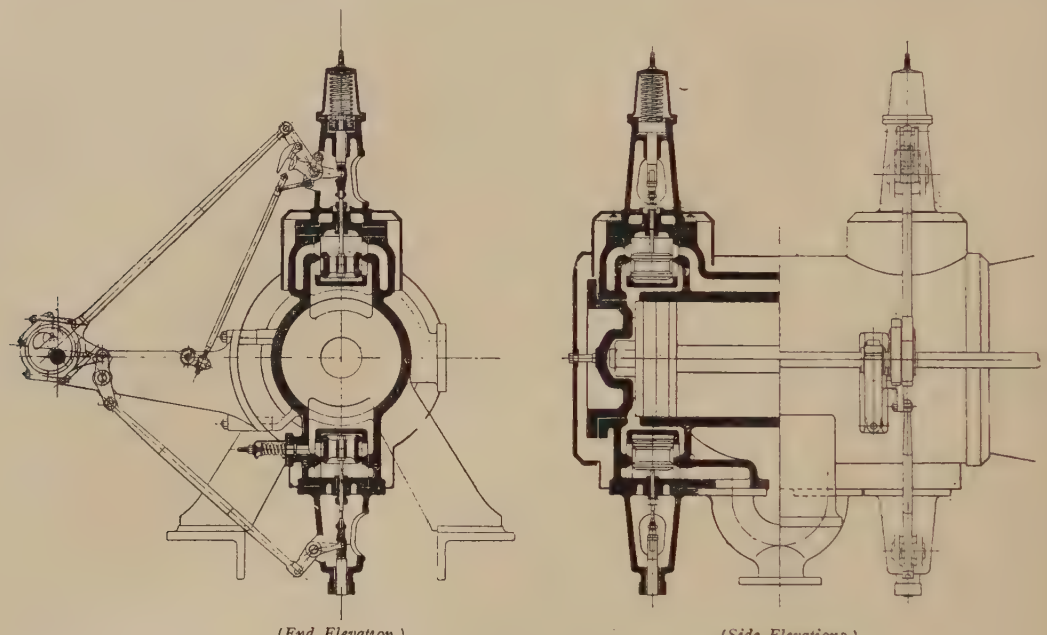
FIG. 34.—Diagram showing the similarity between the " D " Slide Valve and the Piston Valve.

in many triple-expansion engines we find that piston valves are fitted to the high- and intermediate-pressure cylinders and flat valves to the low-pressure cylinder. Both of these types of valve are, however, at a disadvantage compared with the mushroom valve which, as we have seen in Volume II of this work, has been developed for use in the internal combustion engine. The great advantage of the mushroom valve lies in the fact that it only moves sharply to open and shut when required, while the slide valve is always in motion backwards and forwards. This is a serious disadvantage, particularly in slow-speed engines, leading to considerable loss of efficiency, and thus we find that in the modern slow-speed reciprocating steam engine the slide valve has given place to valves of the mushroom type operated from a side shaft in a manner not dissimilar to that found on large gas and oil engines.

A further development is also to be noted. Where slide valves are employed, the steam enters the cylinder and leaves it after having done its work through the same ports. The exhaust steam is, however, at a lower pressure and also at a lower temperature than the incoming steam, so that at every stroke of the piston the ports are chilled by the outgoing steam. Hence the next time the steam enters the cylinder a certain proportion of its heat is wasted in raising the temperature of the ports, and this, in addition to being a cause of loss of efficiency, means that condensation occurs leading to the accumulation of water in the cylinder with all its attendant evils. If, however, an arrangement could be provided which would give separate inlet and outlet passages for the steam, much of this loss would be eliminated. The internal combustion engine with its separate inlet and exhaust valves probably supplied the key to the solution of this problem ; in any case, we find that separate steam and exhaust

valves working in separate passages are provided in these high-powered, horizontal reciprocating steam engines, the engines being generally known as " drop valve " engines. The result of these two improvements, viz. the use of mushroom-type valves and the provision of separate steam and exhaust passages, has been to yield a far simpler and more compact cylinder for a given power as compared with the type having a slide valve working in a separate steam chest, while the efficiency of the engine has been substantially improved.

As an example of this type of reciprocating steam engine, we may consider the engine shown in Plate I, which is a reproduction of a photograph of a horizontal " drop valve " engine by Messrs. Galloways, Ltd., of the Knott Mill Iron Works, Manchester. This engine is of the " back-pressure " type, being designed to run on high-pressure steam, the exhaust, at a pressure slightly above atmospheric, being used for heating purposes in the



(*End Elevation.*) (*Side Elevations.*)

FIG. 35.—General Arrangement of Cylinder and Valve Gear of Galloway Drop Valve Engine.

works. A transverse and half-longitudinal section through the cylinder of an engine of this type is shown in Fig. 35, from which it will be seen that separate steam and exhaust valves are provided at each end of the cylinder, the admission valves being at the top and the exhaust valves at the bottom. This arrangement is generally adopted because it enables any water formed in the cylinder to automatically drain away, and it gives a short and simple connection for the exhaust to the condenser which is frequently arranged immediately beneath the engine. It will be noticed also that while all the valves are of cylindrical form, a departure has been made from the usual mushroom form in favour of a " double-ported " valve in which, since the central portion of the disc is removed, the steam pressure on both sides is practically balanced and thus the force required to raise the valves is reduced to a minimum. The valves are made of cast iron, and are fitted with duplex piston rings made by a special process developed by Messrs. Galloways. The valve liners are also made of hard cast iron, and are forced into the patent cylindrical valve chambers which are cast on the cylinders. The general form of the valve and its liner is shown in the end elevation. It should be noticed also that both the

valve and the liner are, when working, surrounded by steam, so that any variations of temperature will affect both to an equal extent and thus the circularity of the liner and, with it, its steam-tightness are preserved.

The steam valves are operated by an improved form of trip gear operated by means of oil pressure, the point of cut-off being controlled by the governor. The general arrangement of the trip gear is shown in section in Fig. 36, while the appearance of the gear can be gathered from the photograph reproduced on Plate I. Oil under pressure is applied below a plunger A connected to the valve spindle B. Thus the movement of the plunger lifts the valve, thereby admitting steam to the cylinder. The valve closes at the correct instant by the action of a spring and dashpot C, when the pressure of the oil is released. An eccentric mounted on the lay shaft of the engine operates the ram D of the oil pump; the ram, which is hollow, has a ring of slots or ports E, and these ports at certain parts of the travel of the ram will be covered by the sleeve F surrounding the ram, the axial position of this sleeve being controlled by the governor as shown. On the up stroke of the pump, oil is forced into the pressure chamber below the plunger, thus raising the steam valve. At a point determined by the position of the governor, the ports in the hollow pump ram overrun the sleeve and the oil escapes through the ports back into an oil reservoir. The steam valve is now free to close under the combined action of the spring and the dashpot piston, the seating of the valve being both rapidly and quietly performed. Trip gear is fitted to slow-speed engines in order to give a far more rapid cut-off than is possible with a slide valve moving at comparatively slow speed over the ports, it being found that a quick cut-off prevents what is termed "wiredrawing" of the steam, i.e. a gradual falling off of pressure due to the narrowing of the opening through which the steam passes before the actual point of cut-off is reached. This wiredrawing must, of course, mean a reduction in the amount of steam entering the cylinder, with consequent loss of power. The movement of the exhaust valve is not of such great importance as that of the steam valve, although it is desirable that the motion at the point of opening should be as quick as possible. In this engine the exhaust valves are actuated by an eccentric through a double-ended lever with a toggle action, as shown in Fig. 35, which gives a prompt opening and closing of the exhaust valve.

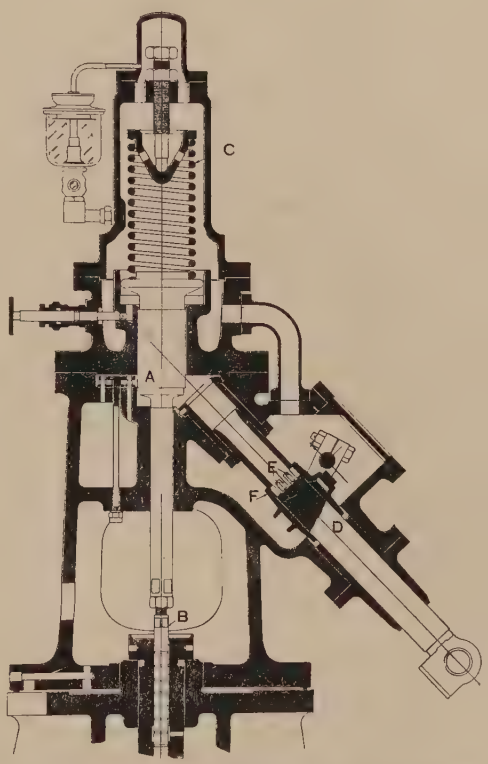


FIG. 36.—Oil-pressure Trip Gear for Galloway Drop Valve Engine.

It will be noticed from the illustration of the engine (Plate I) that the lay shaft follows gas engine practice very closely. The shaft is driven by gearing running in an oil bath, although with the steam engine the speed of the lay shaft must be the same as that of the crank shaft instead of one-half, as is the case with gas engines. The shaft is supported in ring-oiled bearings, and in addition to operating the valve rod eccentrics it also carries the gear for driving the governor. In addition, a simple

form of eccentric is employed to operate the lubricating oil pump. It may also be mentioned that the crosshead guides are of the trunk pattern forming part of the engine frame and are bored out at the same time as the flange, which is bolted to the cylinder cover, thus ensuring correct alignment of the engine. The piston is also provided with a tail rod, which is carried by a slide, thus relieving the cylinder and the cylinder cover stuffing boxes of the weight of the piston, reducing wear of the packing and preventing leakage of steam either from one side of the piston to the other or from the cylinder through the stuffing boxes to the engine room. The crank is enclosed and the big end bearing is lubricated under pressure, as also are all the other bearings, thus ensuring smooth and continuous running.

Reciprocating engines of this type are particularly suitable for use in works where large quantities of steam at comparatively low pressure are required for heating purposes, as in paper mills, bleach works, dye works, flour mills, etc. A single-cylinder engine, such as is shown in Plate I, can be used for this work if the load on the engine

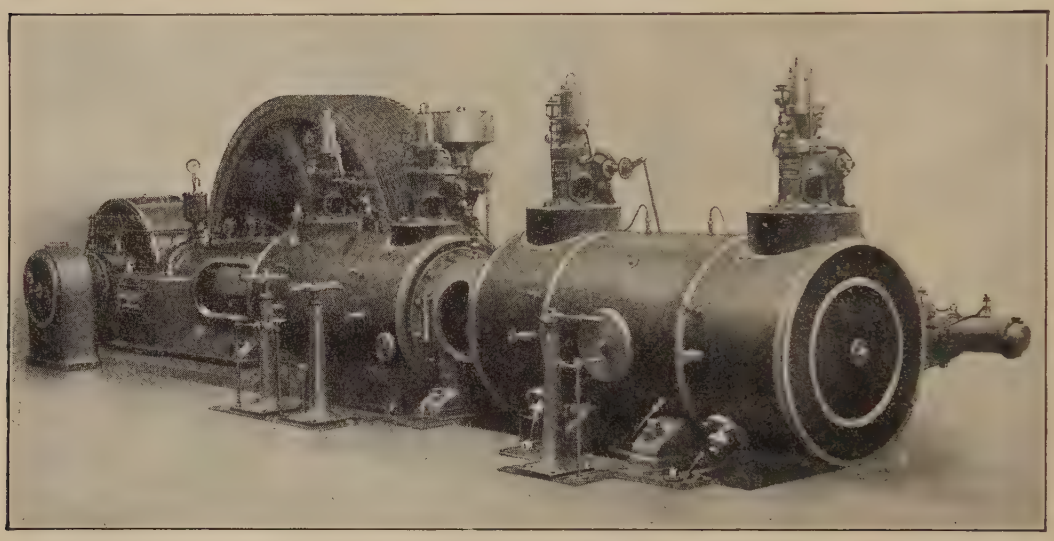


Fig. 37.—Galloway Tandem Compound Drop Valve Extraction Engine of 350 max. i.h.p. arranged for Rope Drive.

is nearly constant and the demand for steam is equal to the total amount passing through the engine. Where, however, the demand for heating steam and the power requirements are liable to fluctuate, other arrangements are necessary, and Messrs. Galloways have evolved a special type of tandem compound steam "extraction" or "pass-out" engine fitted with automatic devices which ensure a steady pressure of the heating steam as well as a constant engine speed, despite varying conditions of load on the engine and changes in the demand for heating steam. In this engine there is a high-pressure and a low-pressure cylinder, these being arranged in tandem, i.e. with their centre lines in the same straight line, and having a common piston rod. The heating steam is drawn from the low-pressure "receiver," where the exhaust from the high-pressure cylinder enters the low-pressure admission valve chests. The high-pressure cylinder takes the place of a reducing valve, so that the steam is doing useful work while its pressure is being reduced from boiler pressure to the required pressure in the receiver of the low-pressure cylinder. Such steam as is not required for heating purposes passes direct into the low-pressure cylinder, expanding and doing further work before it finally exhausts into the condenser. In this way the net steam consumption per indicated horse power

per hour of the engine is reduced to a very small figure, the power being in effect obtained as a by-product of the heating steam.

An engine of this type suitable for use in a small factory is shown in Fig. 37. This engine, which is employed for rope driving, has a normal rated output of 275 i.h.p. with a maximum of 350 i.h.p. The steam pressure at the stop valve is 150 lb. per sq. in., the steam being superheated 150° F. The engine works in conjunction with a condenser, having a vacuum of 26 to 27 in. For heating purposes steam is extracted from the low-pressure receiver at a pressure of 28 lb. per sq. in. In order to maintain automatically a steady pressure in the pipe supplying steam to the factory for heating purposes, a " pressure regulator " is fitted which acts by varying the cut-off in the low-pressure cylinder in accordance with the pressure in the receiver. When the amount of steam required for heating purposes is a maximum, very little steam is allowed to pass into the low-pressure cylinder, almost all the steam which is exhausted from the high-pressure cylinder being taken away as " extraction " steam into the factory steam main, the pressure regulator having brought the low-pressure valve gear to its earliest cut-off. Thus almost all the power of the engine is being developed in the high-pressure cylinder. If now the demand for heating steam decreases, the pressure regulator alters the cut-off of the low-pressure cylinder, making it later so that more power is now developed in this cylinder. This would tend to make the speed of the engine increase if there was not a corresponding increase in the load on the engine, but any tendency towards increased speed is effectively checked by the centrifugal governor which controls the cut-off to the high-pressure cylinder. The combined effect of the governor is, therefore, to limit the change of speed, to limit the change of pressure in the receiver pipe, and to transfer a part of the mechanical load from one cylinder to the other. It may be mentioned that when the demand for steam in the factory is greater than that which it is intended should pass through the engine or when the engine is standing idle, steam is allowed to pass directly from the main steam pipe through a reducing valve into the low-pressure main. On the other hand, if no steam is required for heating purposes, the engine is run as an ordinary compound condensing engine.

A very good idea of the extremely economical performance of these engines can be gathered from the following table, which gives the steam consumption of an engine similar to that shown in Fig. 37, at various loads, when running as a condensing engine, and with different amounts of steam extracted from the receiver. The consumptions given are the amounts condensed per i.h.p. per hour.

STEAM CONSUMPTION OF GALLOWAY TANDEM EXTRACTION ENGINE AT VARIOUS LOADS OR FOR VARIOUS AMOUNTS OF STEAM EXTRACTED FROM THE RECEIVER

I.h.p. developed by engine	130	200	275	300
When passing out no steam: amount of steam condensed, i.e. net steam consumption, lb. per i.h.p. per hour	12·7	11·7	11·4	11·9
When passing out 1,500 lb. of steam per hour: amount of steam condensed, i.e. net steam consumption, lb. per i.h.p. per hour	7·7	8	8·5	9·3
When passing out 3,000 lb. of steam per hour: amount of steam condensed, i.e. net steam consumption, lb. per i.h.p. per hour	—	5·4	6·3	7·3

It will be noticed that when passing 3,000 lb. of steam per hour to the heating system at 130 i.h.p. no steam passes to the condenser; in other words, this power is wholly developed in the high-pressure cylinder. Attention may also be called to the fact that when passing no steam, i.e. when the unit is working as a normal tandem compound engine, at 275 i.h.p. the steam consumption is 11·4 lb. per i.h.p. per hour.

This is an extremely satisfactory performance and one which compares favourably with the performance of many steam turbines.

One of the most famous firms in Great Britain which specialises in the construction of reciprocating steam engines is Messrs. Robey & Co., Ltd., of Lincoln, and among the different engines built by them there are units capable of developing powers up to 10,000 h.p. In order to develop such large powers as these, horizontal engines working either compound or triple expansion with the cylinders arranged tandem or with two engines coupled together side by side are generally employed, and the units are used not only for driving the machinery in large industrial establishments, but for driving air compressors such as are required in the operating of blast furnaces for the production of iron. They are also widely employed for driving the winding gear for coal mines. The drop-valve type of engine as designed and constructed by Messrs. Robey contains many un que features, and it will be of interest to consider briefly the details of these engines. As an example of a large drop-valve engine we may consider the unit shown in Fig. 38. This is a twin tandem compound engine, there being two exactly similar engines coupled on either side of a grooved flywheel for driving machinery by means of ropes. The unit shown has two high-pressure cylinders, each 24 in. diameter, two low-pressure cylinders, each 40 in. diameter, the stroke of the piston in all cylinders being 48 in. The engine develops about 1,500 i.h.p. at a speed of 75 revs. per min. when supplied with steam at a

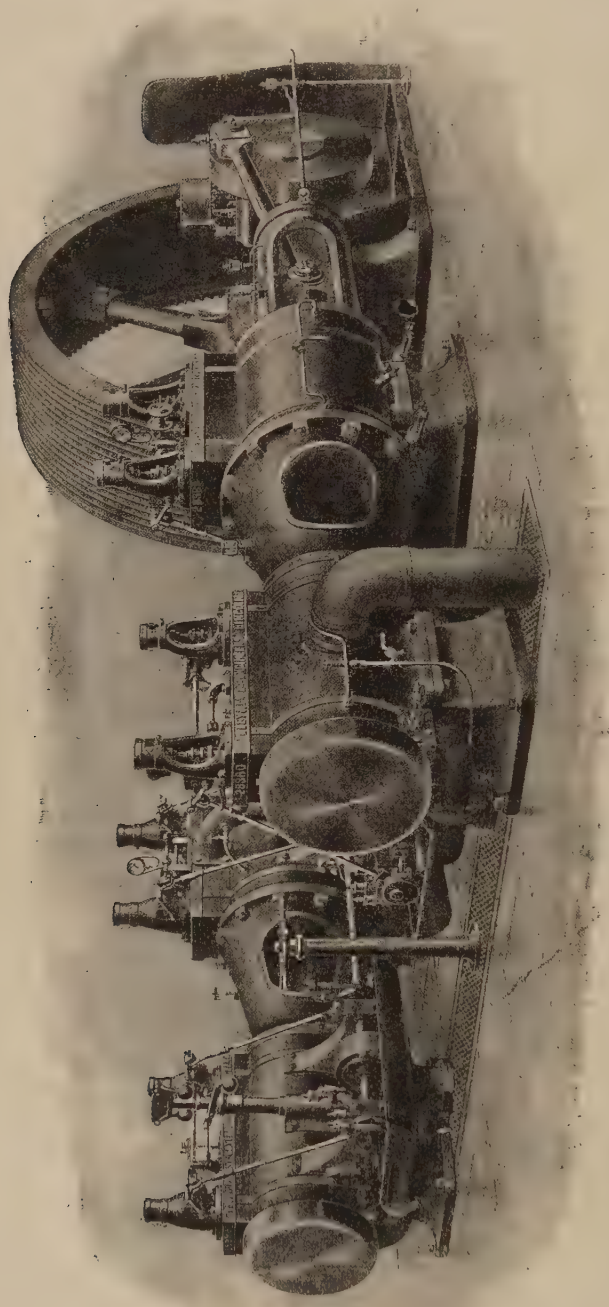


Fig. 38.—Twin Tandem Compound Engine developing 1,500 i.h.p. at 75 revs. per min., designed and constructed by Messrs. Robey & Co., Ltd., Lincoln.

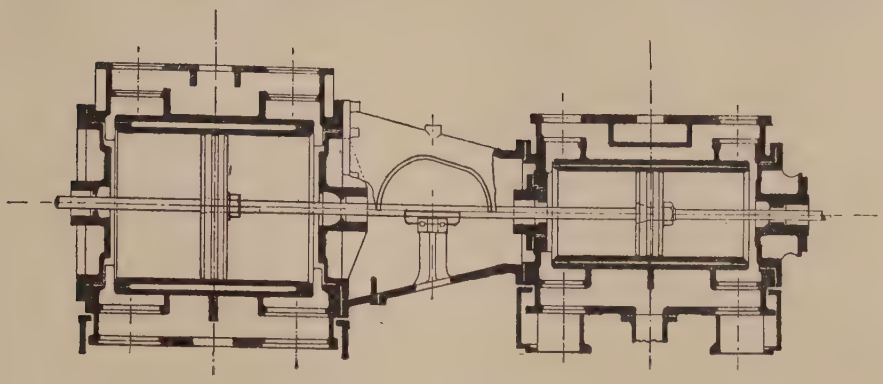


FIG. 39.—Section through Cylinders of Robey Horizontal Tandem Drop Valve Engine.

pressure of 125 lb. per sq. in. and running condensing, i.e. not as a back-pressure engine, but of course higher powers would be obtained if the engine is designed to run on steam at a higher pressure. The power mentioned is that which the engine can develop with the maximum economy, but it should be noted that the engine is capable of developing a maximum of about 1,750 i.h.p., although this latter figure is only obtained at a slightly higher steam consumption. A good idea of the general overall dimensions of the engine will be gathered from the fact that the flywheel is 20 ft. diameter.

A section through the cylinders constituting one line of this engine is shown in Fig. 39, from which it will be noticed that double-ported valves are employed, both for the admission of the steam and for the exhaust. Each cylinder is provided with a detachable liner, which can be renewed if desired with the minimum of trouble and expense without adversely affecting other parts of the engine. An enlarged sectional end elevation through one of the cylinders taken through the valves is shown in Fig. 40, and in this drawing the general arrangement of the valve-operating mechanism is also shown. It

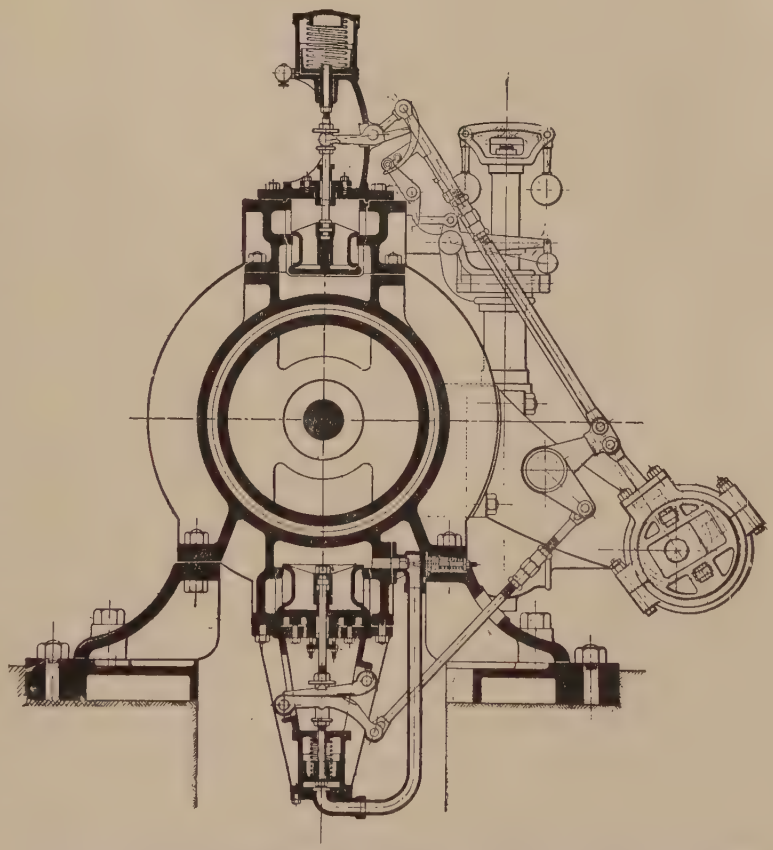


FIG. 40.—Sectional End Elevation through Cylinder and Valves of Robey Drop Valve Engine.

will be noticed that the gear is actuated from a lay shaft running along the side of the engine with its axis parallel to the common axis of the tandem cylinders. This lay shaft is driven by means of skew gearing from the main crank shaft, its speed being the same as that of the crank shaft. Both the admission and the exhaust valves are operated by the same eccentric, the admission valve which is above the cylinder being actuated directly by the eccentric rod through a rocking lever, while the exhaust valve underneath the cylinder is operated through a bell crank lever, one end of this lever being connected to the eccentric rod, while the other is connected to the lever, which actuates the exhaust valve through another rocking lever. The admission valve is not opened directly by the motion of the eccentric rod, its motion being controlled by means of the "Robey" trip motion, which gives that sharp motion to the valve at the points of admission and cut-off of the steam which is so essential if wire-drawing of the steam is to be prevented.

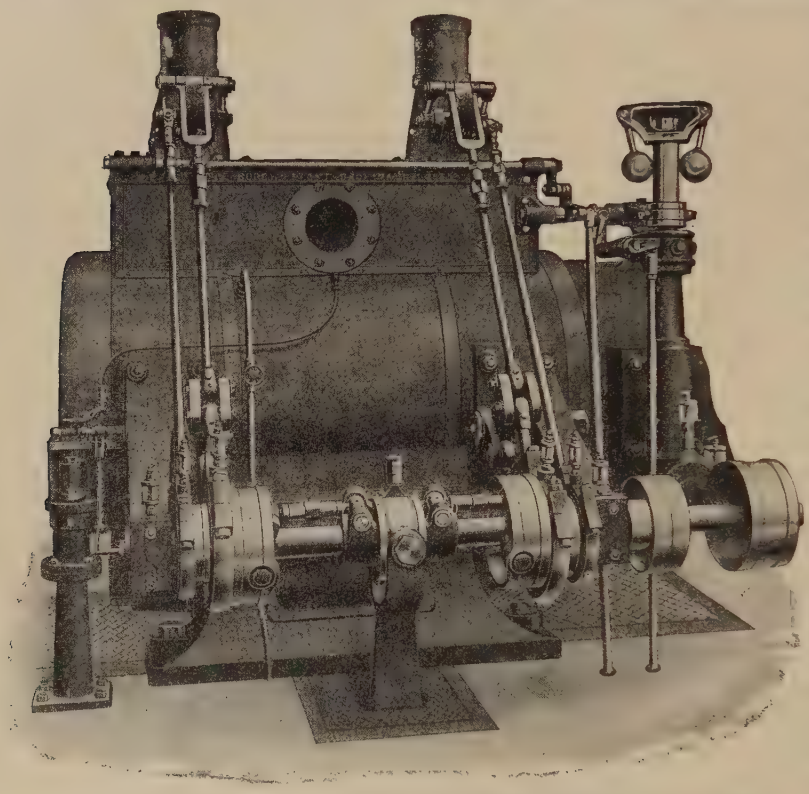


FIG. 41.—Cylinder of Robey Drop Valve Engine, showing Valve Gear, Governor Gear, and Reversing Gear.

In the majority of engines which we have so far studied it has not been necessary to provide the engine with any form of reversing gear, since it is only required to run the engine in one direction. Thus for driving electrical generators, pumps, or air compressors, in fact for the majority of industrial purposes, the main engine only runs in one direction and therefore it is not fitted with any mechanism which will enable motion in the reverse direction to be obtained. But in certain circumstances, such as when driving pit-head winding gear or steel rolling mills, motion in both directions is absolutely essential, and hence the engine must be fitted with an efficient form of reversing gear. A very simple but efficient form of reversing gear has, therefore, been designed which is fitted on these engines in all circumstances where motion in either direction is required. Referring again to Fig. 40, it will be noticed that the eccentric which operates the valves is driven by means of a rectangular block secured to the lay shaft and that the eccentric sheave is provided with a long rectangular-shaped hole in which this block works. In order to reverse the engine, all that is necessary is to shift the eccentric sheave in a direction at right angles to the eccentric rod, so that the block moves relatively to the sheave from one side of the hole to the other. This changes the moment at which the

valves open and close relatively to the position of the pistons and thus causes the engine to run in the opposite direction when steam is admitted. A corresponding motion must of course be given to each eccentric sheave at the same time.

In order to slide the eccentric sheaves along the rectangular blocks, a sliding sleeve is mounted on the lay shaft midway between the two eccentrics, and to each end of this sleeve is fitted a rod which at its other end is connected to a bell crank lever. Each of these levers is pivoted to a small drum carried on the lay shaft, and the other ends of the levers are secured to the eccentric sheaves, the whole of the gear from the sleeve to the bell crank levers rotating with the lay shaft. Thus a motion of the sleeve along the lay shaft is transmitted to the eccentric sheaves, causing them to slide along the rectangular blocks and thus reversing the motion of the engine. The sliding sleeve is operated by means of a fork which is connected to the reversing lever of the engine, and so small is the force required to slide the eccentrics that, except in the larger size of engines reversing is carried out by hand. The arrangement of the reversing gear together with the forked lever which actuates it is shown in Fig. 41, which is a reproduction of a photograph of a complete cylinder with its valve gear and reversing mechanism.

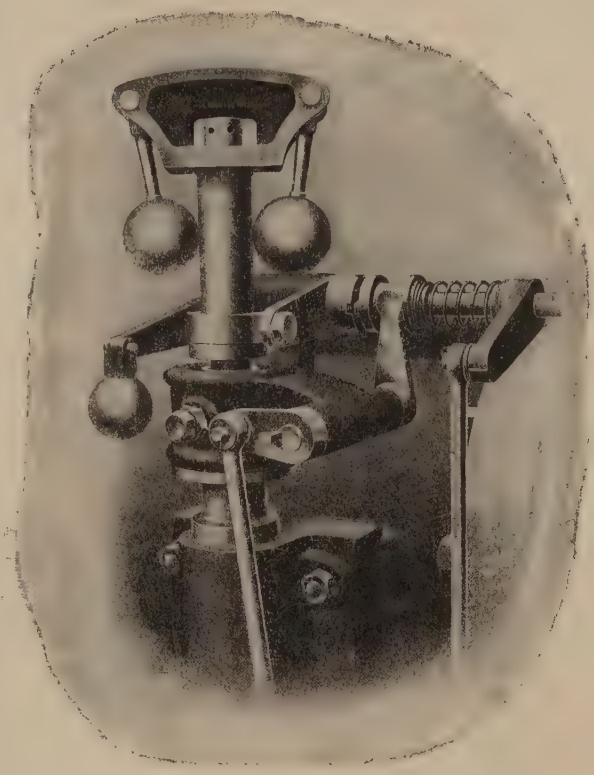


FIG. 42.—Governor for Robey Drop Valve Engine.

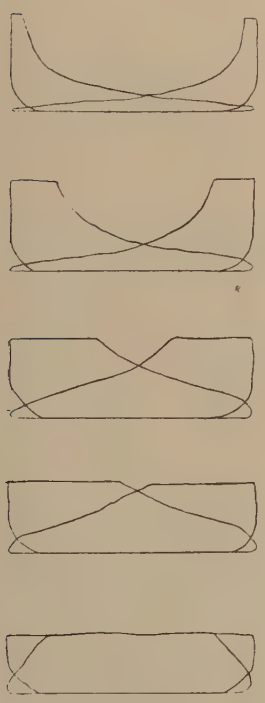


FIG. 43.—Indicator Diagrams from Robey Drop Valve Engine, showing Range of Cut-off.

The governor of the engine which is shown in Fig. 42 is of the centrifugal type, which controls the point of cut-off of the steam. Where a wide range of control is required, as, for example, with winding engines, a second eccentric is keyed on to the lay shaft in a fixed position which does not vary with the running of the engine. These second eccentrics can be readily seen at the side of the main eccentrics in Fig. 41. The motion of the admission valve is controlled by a rocking lever which is itself operated by a curved tripper, this tripper being moved by the main eccentric rod and tripped by a roller which makes contact at the bottom part of the curve. This roller is moved by the rod actuated by the auxiliary eccentric, the upper end of this rod being secured to the lever keyed on a small lay shaft, the exact position of which is controlled by the governor. When the balls of the governor

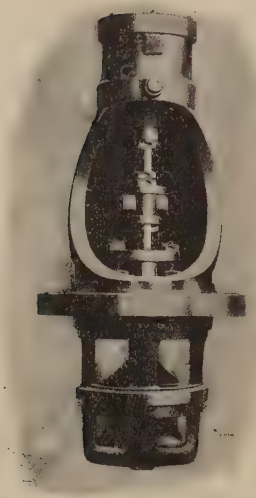


FIG. 44.—Double-ported Valve and Cage for Robey Drop Valve Engine.

are in their lowest position, the cut-off is at the maximum position, which is at about 90 per cent. of the stroke. As the speed of the engine increases, the balls tend to fly outward, thus moving the roller and causing the valve to close earlier in the stroke. When maximum speed has been obtained, the cut-off is at its earliest point, which is at about 5 per cent. of the stroke. Some idea of the wide range of control obtained with this governor gear will be gathered from the series of indicator diagrams which are reproduced in Fig. 43. These have been taken for both sides of the piston on the same diagram, and show that, despite the wide variation in the point of cut-off, the conditions at all other points on the diagram, viz. the point of admission of steam and the opening and closing of the exhaust valve, are not affected.

The steam admission valve together with its housing is shown in Fig. 44, from which it will be seen that the valve spindle is provided with two collars, the upper one being adjustable, and between these the forked end of the rocking lever which operates the valve is fitted. A dashpot is mounted above the valve spindle, so that its motion while quick is silent and free from shock. The valve seating forms a cage which is entirely separate from the cylinder casting, from which it can be easily removed for grinding the valve seatings or for renewal. The valve seats are provided with soft copper rings secured in such a way that no jointing material is required. The valve spindle works through a stuffing box provided with metallic packing.

For units of the largest power, the triple-expansion principle is employed, the high- and the intermediate-pressure cylinders being arranged in tandem driving one crank, with the low-pressure cylinder driving another crank, a heavy flywheel being mounted on the shaft between the cranks. One of these engines is shown in Fig. 45, the h.p. and

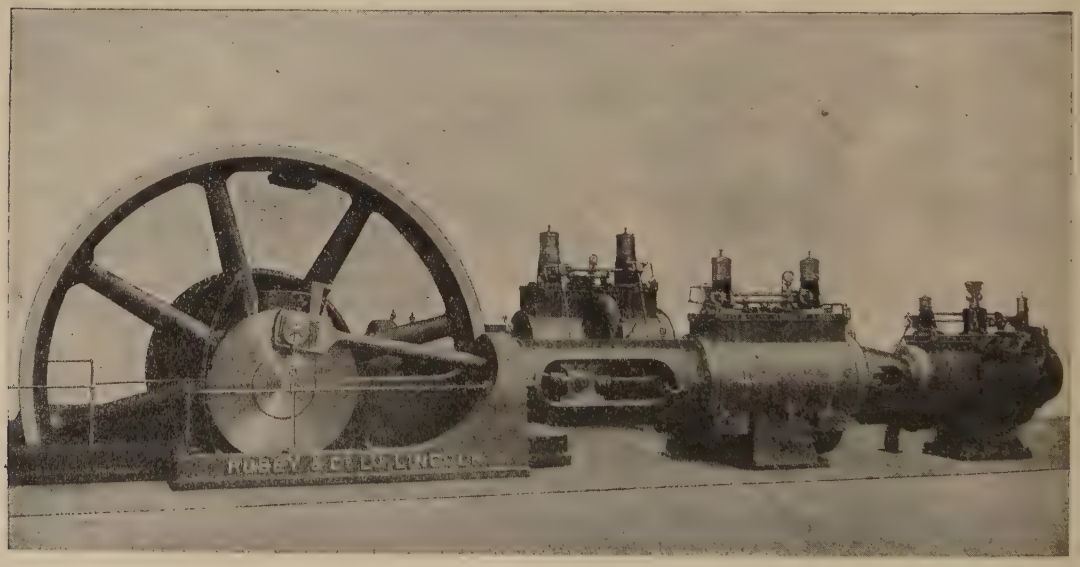


FIG. 45.—Robey Triple-expansion Two-crank Drop Valve Engine. Engines of this type are made up to 10,000 h.p. per unit.

i.p. cylinders being at the front. These engines are particularly suitable for driving electrical generators owing to the even turning moment provided by the two cranks assisted by the heavy flywheel, and as might be expected where triple expansion is employed, the efficiency of the engine is high. This design is adopted for engines developing up to 10,000 horse power, i.e. capable of driving generators having an output of about 7,000 kw.

One of the most important applications of the Robey " drop-valve " engine has been for the driving of pit-head winding gear. For this purpose a simple but efficient form of reversing gear is absolutely essential and in addition some form of overwinding and overspeed prevention gear must also be fitted. This gear not only renders it impossible to overwind the cages, but it prevents too high a speed when the cages are reaching the surface, although it can be arranged to prevent overspeed at any point in the shaft. In addition it prevents the starting of the engine in the wrong direction. A good idea of the appearance of one of these engines can be obtained from Fig. 46, which shows a drop valve engine driving directly a semi-conical winding drum. This engine consists of two single-cylinder non-compound units, the pistons having a stroke of 72 in. A separate view of the winding drum mounted on the crank shaft is shown in Fig. 47, and it may be noted that the cranks are arranged at right angles to each other, thus enabling the engine to be started up in any position. The winding drum is 24 ft. diameter at its largest part and 14 ft. diameter at the smallest part of the cone, while the general proportion of the various parts will be realised by comparison with the man on the right.

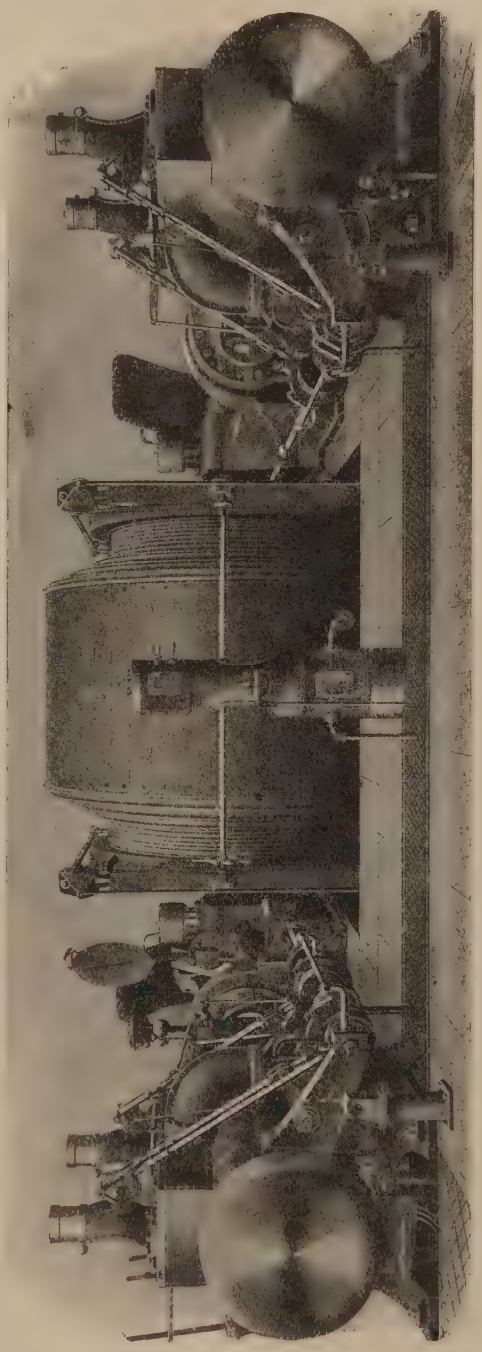


Fig. 46.—Robey Twin Drop Valve Engine driving Pit-head Winding Gear.

A closer view of the Robey patent overwinding and overspeed prevention gear is shown in Fig. 48, from which it will be seen that the gear is mounted on the engine lay shaft, i.e. on the shaft which carries the eccentrics operating the valves, being actuated by a nut which moves along the lay shaft as the latter rotates, the position of this nut corresponding to the position of the cages in the shaft. An auxiliary governor is mounted on the lay shaft nearer the engine, as shown, and by

means of a bell crank lever, the movement of the balls of this governor imparts an angular motion to an auxiliary shaft, which is mounted parallel to the lay shaft. This auxiliary shaft carries two stepped sleeves, and if the engine is running at too high a speed, these sleeves come into contact with the traversing nut before the end of the wind, thus putting into action the mechanism connected to the governor, which has the effect of slowing down the speed of the engine. At the same time another mechanism is called into play by the contact with the stepped sleeves, which applies the brake. The speed at which it is desired to operate the overspeed gear can be adjusted as desired. If the speed has been duly reduced by the driver as the cages approach the bank, the overspeed device will not come into action. It might happen, however, that instead of

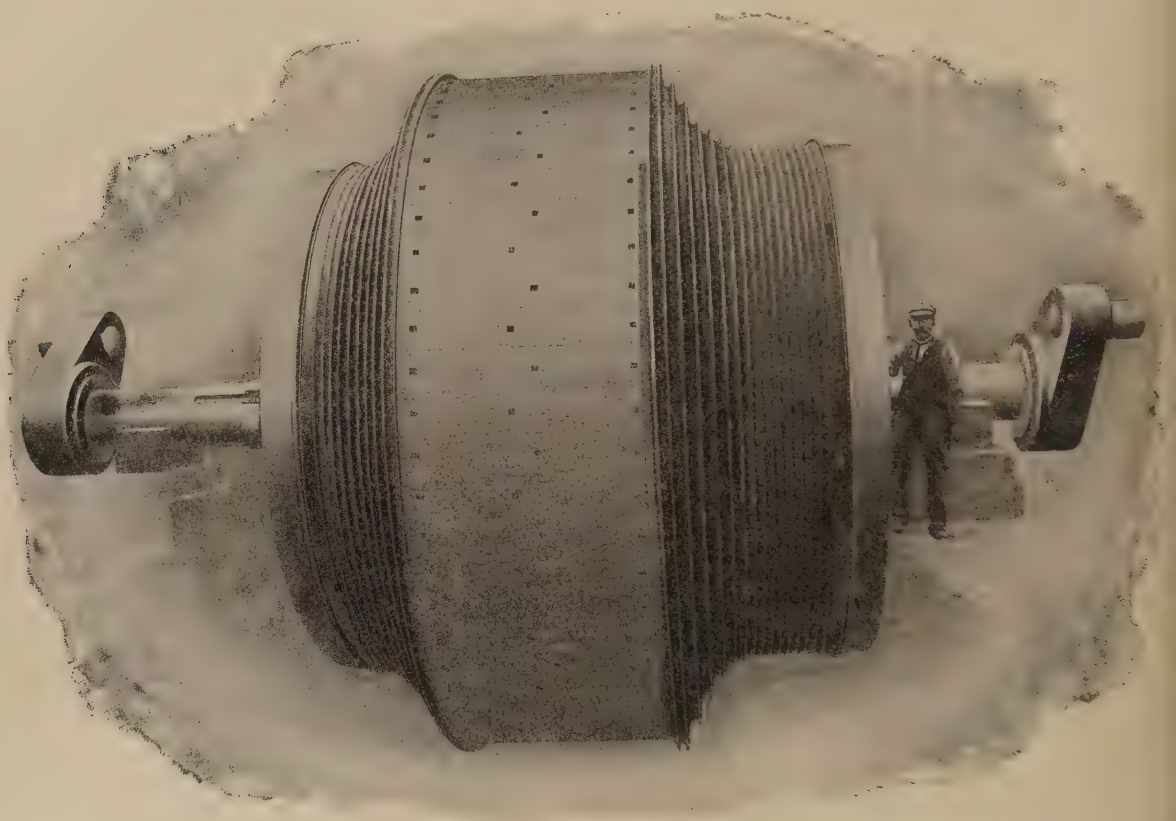


FIG. 47.—Winding Drum mounted on Crank Shaft of Robey Twin Drop Valve Engine.

stopping the engine when the cages reach the bank, he permits it to run on at its reduced speed. In this case a projection on the traversing nut comes into contact with a lever which through a suitable mechanism operates the governor and applies the brake. Thus the cages will be brought to rest in the correct position, no matter whether the engine has been slowed down by the driver or by the overspeed device just described. Further, should the engine driver attempt to start the engine in the wrong direction, the projection just referred to comes into contact with the lever which shuts off the steam and applies the brake. It may be noted that this gear can be applied to existing engines without touching the lay shaft, a self-contained mechanism driven from the lay shaft being mounted alongside the engine.

As has already been stated, the drop valve engine possesses many distinct advantages

over the older slide valve type, but in common with that type, it possesses a great disadvantage, due to the fact that the direction of the steam in the cylinder is continually changing—that is, the steam enters at the end of the cylinder, drives the piston forward, and then during the return stroke leaves the cylinder again at the same end through which it entered. But, as the steam expands driving the piston forward, its pressure falls and this is accompanied by a fall of temperature, so that it follows that each end of the cylinder is alternately heated and cooled by the hot incoming and the cooler outgoing steam. In the slide valve engine the same passages connecting the steam chest and the cylinder are employed both as admission and exhaust ports, as will be noticed on referring to Fig. 13, which shows the section through the cylinder and valve chest

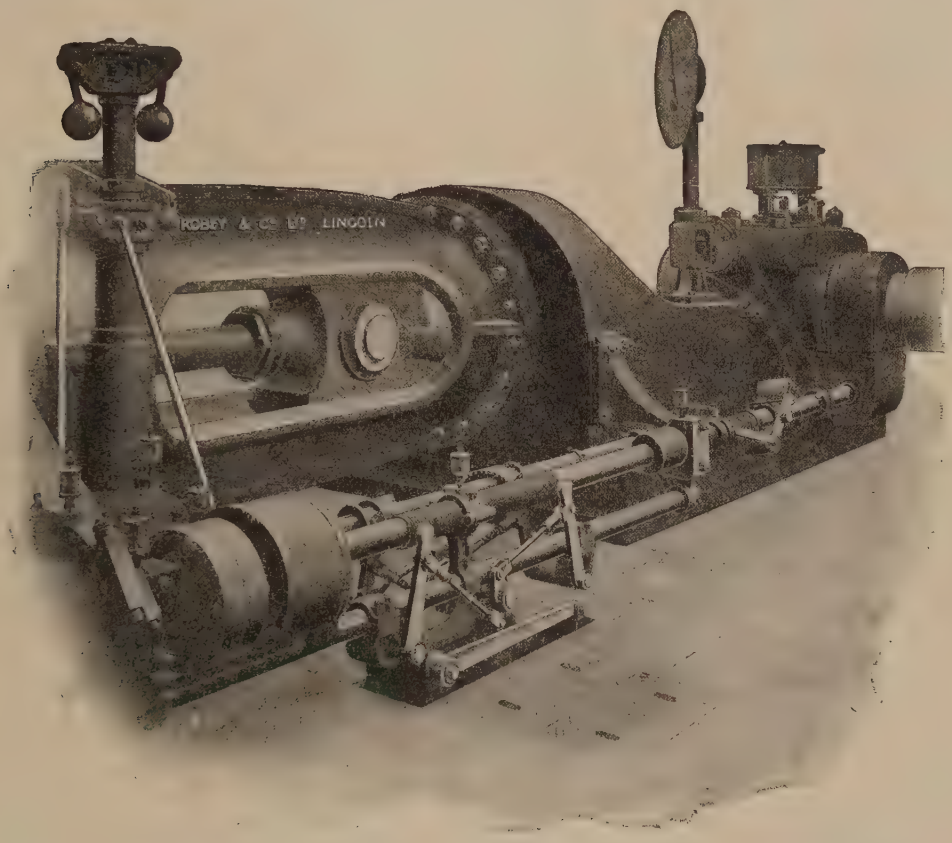


FIG. 48.—Robey Patent Overwinding and Overspeed Device as fitted to Winding Egnines.

of an engine of this type. In the drop valve engine and in other large engines of modern design separate admission and exhaust valves are emp'oyed, but even then the inlet and outlet ports are close together, and with the exhaust steam returning to the same end of the cylinder as that by which it entered, the inner surface of the cylinder cover and the end of the cylinder itself are very much reduced in temperature. Hence, when next the steam enters the cylinder it comes first of all into contact with these cold surfaces, and as a result a large amount of heat is abstracted from the steam before the temperature of these surfaces approximates again to that of the steam. This abstraction of heat results in a considerable amount of condensation in the cylinder, which, apart altogether from the loss of efficiency involved, leads to the many troubles which inevitably follow the accumulation of water in an engine.

To overcome this disadvantage, the " uniflow " type of engine was evolved, in which, as its name implies, the steam flows in one direction only in its passage from the outlet valve to the exhaust. A section through the cylinder of a uniflow engine is shown in Fig. 49, from which it will be seen that the inlet valves are placed at each end of the cylinder, while exhaust ports are provided at the centre, the direction of the steam being indicated by the arrows. The length of the cylinder is made slightly more than twice the stroke of the piston, this latter being in the form of a hollow drum of length almost equal to its stroke. The steam, on entering the cylinder, drives the piston forward, the inlet valve closes, and the piston continues to move forward due to the expansion of the steam, until the exhaust ports are uncovered, when the steam, having done its work, passes to the condenser. Thus the outgoing exhaust does not pass anywhere near the hot inlet surfaces and there is, therefore, no appreciable drop in temperature when the hot steam is again admitted to the cylinder. As a result of this, it is found

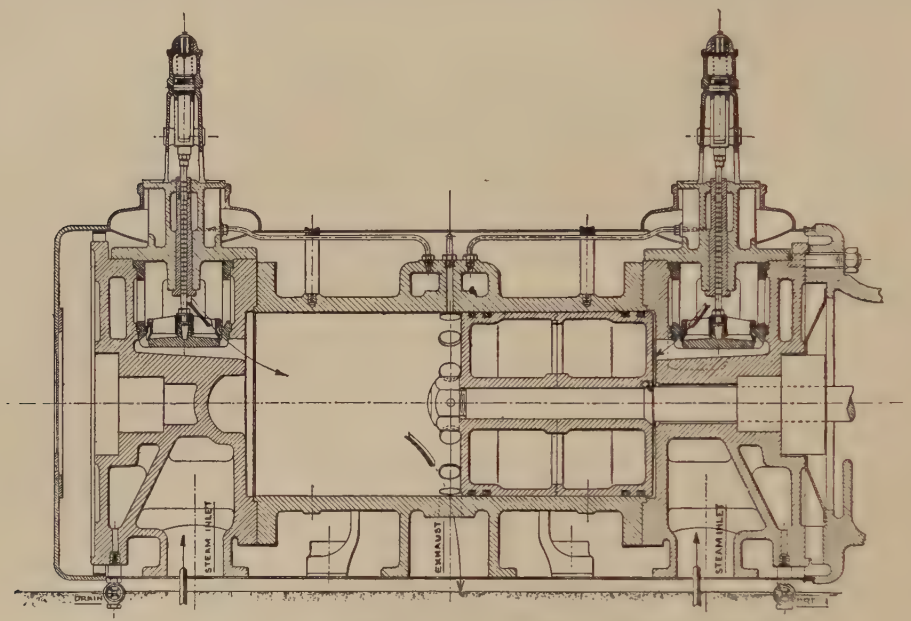
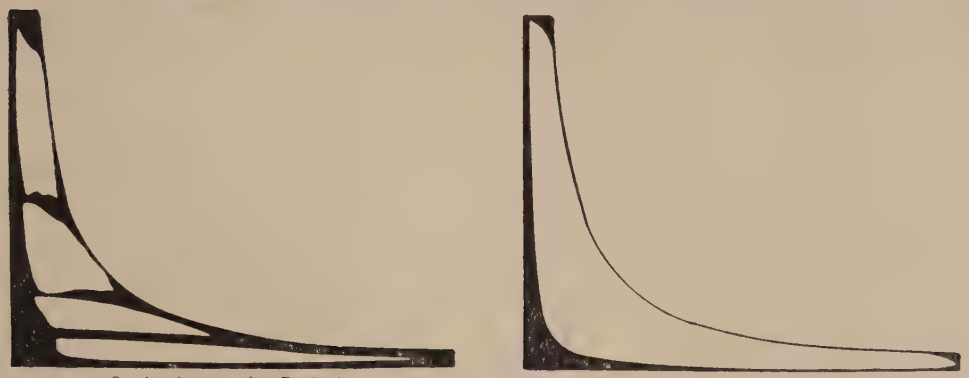


FIG. 49.—Section through Cylinder of Robey " Uniflow " Engine.

that the efficiency of the engine is very high, in fact it can be asserted that the uniflow engine represents the most efficient type of reciprocating steam engine that has yet been evolved.

In addition to the economy which follows the practical elimination of cylinder condensation, due to the employment of the uniflow principle, it has been found practicable to employ a far greater range of expansion in one cylinder than is possible in any other form of reciprocating engine. This is partly due to the fact that with the elimination of the exhaust valve and the ports leading to this valve, the clearance space, i.e. the space available for steam when the piston is right at the end of its stroke, including the volume of the ports together with that between the piston and the cylinder cover, can be made much smaller than is possible when two sets of ports are employed. Further, in the uniflow engine a far wider range of temperature between inlet and exhaust is possible, since the cold exhaust does not pass over the surfaces heated by the hot incoming steam, and this means that the ratio of the inlet pressure to the exhaust pressure, or in other words the ratio of expansion, can be made greater. Hence if the cut-off occurs



(*Quadruple-expansion Engine.*) (*Uniflow Engine.*)

FIG. 50.—Diagrams showing the Comparative Loss in a Quadruple-expansion Engine and a Uniflow Engine. The black area represents the loss in each case.

very early in the stroke, the volume of steam enclosed in the cylinder, i.e. the clearance volume plus the small volume swept through by the piston, is so small that when the piston has just reached the exhaust ports, the increase in volume will be equivalent to a very high ratio of expansion. In fact in the majority of uniflow engines it is found practicable to allow the entire expansion between boiler and condenser to take place in one cylinder, which adds further to the efficiency by eliminating the extra mechanical losses due to the employment of two, three, or even four cylinders, as in the compound, triple-expansion or quadruple-expansion engine. The saving obtained by the use of the uniflow engine is clearly demonstrated by the diagrams reproduced in Fig. 50. On the left are shown the indicator diagrams from a quadruple-expansion engine combined on common axes of pressure and volume to show the continuous expansion of the steam, while on the right to the same scale is the diagram from a uniflow engine. In each case the area shown in black represents the loss, i.e. the energy of the steam which is not indicated as doing work on the pistons. A pair of indicator diagrams taken from a uniflow engine under test are reproduced in Fig. 51, the diagram for the front and the back of the piston being drawn upon one card. The line parallel to the base of the diagram is the atmospheric pressure line, so that it will be seen that expansion occurs from stop valve pressure to an exhaust pressure considerably less than atmospheric.

Tests carried out on uniflow engines of different designs have shown that exceedingly low steam consumption can be obtained with these engines under ordinary commercial operation, and, moreover, it has been proved that this improved economy is obtained not only in very large engines, but also in engines of very small power. In addition to this, the high efficiency is secured not only at the designed load, but over the whole range of load possible for the engine. Under normal conditions, the uniflow engine has a steam consumption of 10 to 11 lb. per i.h.p. per hour, depending, of course, upon

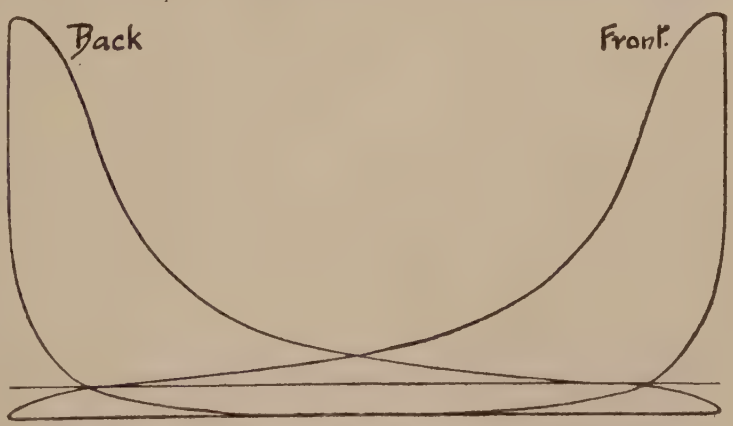


FIG. 51.—Indicator Diagrams from Single-expansion Uniflow Engine.

the initial pressure, the degree of superheat, and the vacuum in the condenser, a figure which is seldom obtained except in large steam turbines.

The pioneers of the uniflow engine in this country were Messrs. Robey & Co., Ltd.,

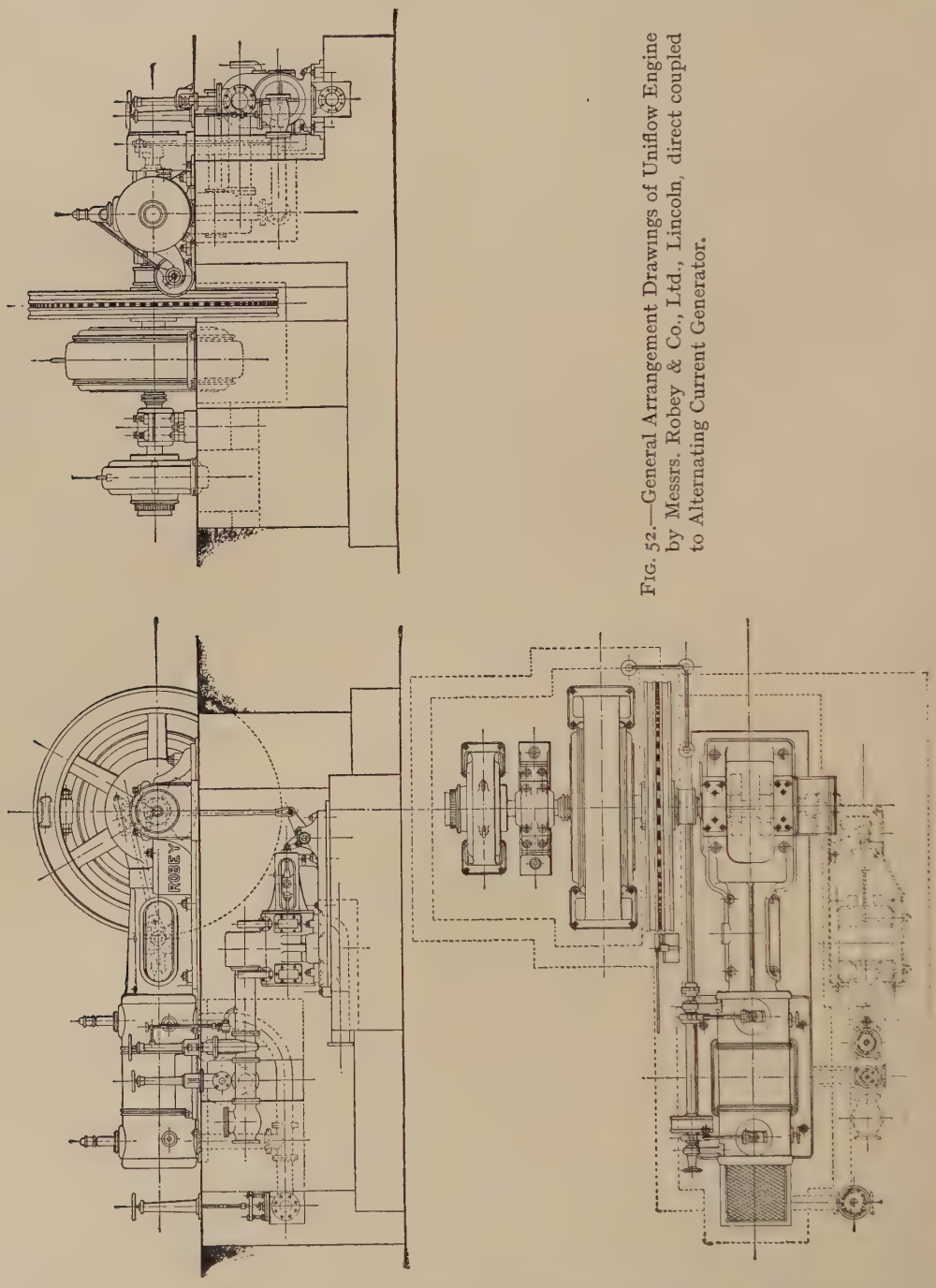


Fig. 52.—General Arrangement Drawings of Uniflow Engine by Messrs. Robey & Co., Ltd., Lincoln, direct coupled to Alternating Current Generator.

of Lincoln, who have built a large proportion of the numerous uniflow engines which are in use in this country. Fig. 49 (page 54) is a section through the cylinder of a Robey engine, and in order to describe in greater detail the design and construction

of this type, we cannot do better than consider the uniflow engine designed and constructed by Messrs. Robey. The general arrangement drawings of one of these engines are reproduced in Fig. 52, the engine being shown direct coupled to an alternating current generator, a small direct current generator for supplying current to the poles being mounted on an extension of the shaft. It will be noticed from the plan that there is a lay shaft driven by the gearing on the main crank shaft, this lay shaft carrying the eccentrics which operate the inlet valves. The lay shaft has the same speed as the crank shaft, and the eccentric rods operate cams which open and close the valves by raising or lowering the rollers carried by bridles attached to the valve spindles. These cams are of special design in order to ensure easy and silent opening and closing of the valves. The governor is also mounted direct on the cam shaft, being of the combined centrifugal and inertia type. The governor operates by turning the lay shaft and thus altering the position of the eccentrics, which in turn causes the point of cut-off of the steam to be varied to suit the load. The governor is provided with speed-adjusting gear which is operated by the hand wheel seen at the end of the lay shaft. The speed can therefore be adjusted while the engine is running.

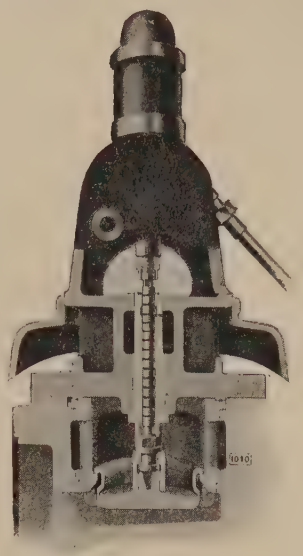


FIG. 53.—Inlet Valve and Housing for Robey Uniflow Engine.

The complete inlet valve together with its seating and the bonnet housing the valve-operating gear is shown in Fig. 53. It will be noticed that the seat is an entirely separate casting from the cylinder and that it is held in position by the base of the bonnet casting. The actual valve seating is provided with a soft copper ring, so that no other jointing material is required. The valve spindle is grooved to form a labyrinth packing[1] in order to prevent leakage of steam and at the same time to reduce friction. The valve is of the double-ported type with two seats, the upper one slightly larger than the lower one in order that the valve can be easily removed if required. The valve is thus practically balanced and the force required to lift it is very small. At the top end of the spindle is a guide plunger working in the upper part of the bonnet cylinder, a spring being provided to ensure that the valve is quickly closed. The valve opens and closes once for every revolution of the engine, and hence the lay shaft must be driven by a "one to one" gear. This gear is shown in Fig. 54, and it will be noticed that it is of the skew type, the number of teeth on the larger wheel which is keyed to the crank shaft being the same as that on the smaller wheel which is keyed to the lay shaft. The smaller wheel works in an oil bath, and, owing to the design of the teeth and the accuracy with which they are cut, the gear is found to run very silently.

FIG. 54.—"One to One" Gear for driving Long Shaft of Robey Uniflow Engine.

The design of the main crank shaft bearings on the Robey engines is of considerable interest. The housing for one of the main bearings is shown in Fig. 55, which is a plan view looking down on the end of the bed plate, the bearing cap being removed. It will be noticed that not only is the bed plate bored out in a direction parallel to the axis of the shaft, but it is

[1] The principle of the labyrinth packing has been fully explained in its application to steam turbine design. See Volume I, p. 66.

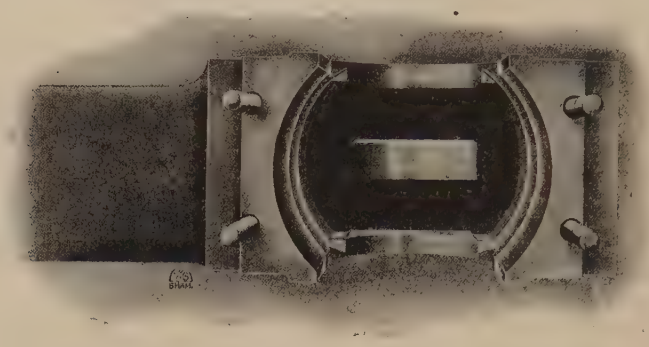


FIG. 55.—Housing for Main Crank Shaft Bearing of Robey Uniflow Engine.

also bored in a vertical direction. The bearing itself is shown in Fig. 56, from which it will be noticed that it is built up in four parts. The top and bottom portions rest on the cylindrical surfaces bored for them in the bed plate and the cap respectively, but the side portions are held in position by two wedges which are shown in position, in Fig. 56, these wedges being planed flat where they come into contact with the bearing, but turned on their outer surfaces to the same diameter as the vertical cylindrical surface in the housing already referred to. Thus while adjustment for wear can always be made by means of the wedges, the circular surface enables the shaft to adjust itself so as to give a uniform pressure all over the side portions of the bearing, these being the portions which take the variable pull and thrust due to the pressure of the steam alternately on the front and the back of the piston. As is to be expected with engines of these high powers, forced lubrication is adopted for all bearings. The oil is delivered direct to the main crank shaft bearings from the pump, then to the crank pin bearing, from which it passes along the connecting rod to the gudgeon pin bearing and to the crosshead guide shoes, the return oil being filtered through brass gauze sieves before passing again to the pump.

In the general arrangement drawing, Fig. 52, it will be noticed that there is a small crank on the extension of the crank shaft. This carries a rod which drives the air pump through a rocking lever. As shown in this drawing, the steam exhausts into a condenser of the jet type, which is placed in close proximity to the engine. The air pump, which is of the double-acting type, draws the condensed steam, as well as any air which may find its way into the system, through a central inlet, delivering it at each end of the pump cylinder into the pipe shown at the bottom of the engine pit in the side elevation. The pump is of simple construction, being fitted with delivery valves only, these being easily accessible through the rectangular doors seen at either end of the pump. It will, of course, be readily understood that any other form of condenser as well as any other form of air pump could be employed, provided they were together capable of efficiently yielding a satisfactory vacuum.

The appearance of these engines as erected will be gathered from Figs. 57 and 58, which are reproduced from two photographs of a small uniflow engine having a single cylinder 29 in. diameter with piston stroke 33 in., developing about 500 b.h.p. at a speed of 140 revs. per min. Steam is supplied at a pressure of 145 lb. per sq. in., superheated to a temperature of 550° F.,

FIG. 56.—Main Bearing for Robey Uniflow Engine.

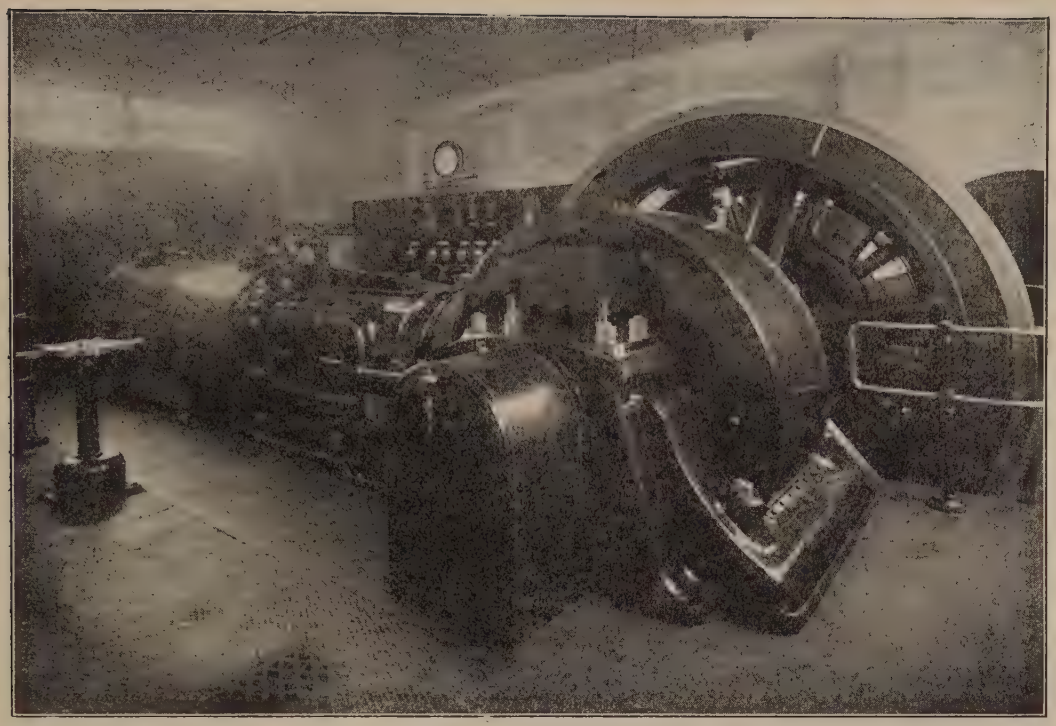


FIG. 57.—Robey Uniflow Engine of 500 b.h.p., driving a 360 kw. Direct Current Generator at 140 revs. per min.

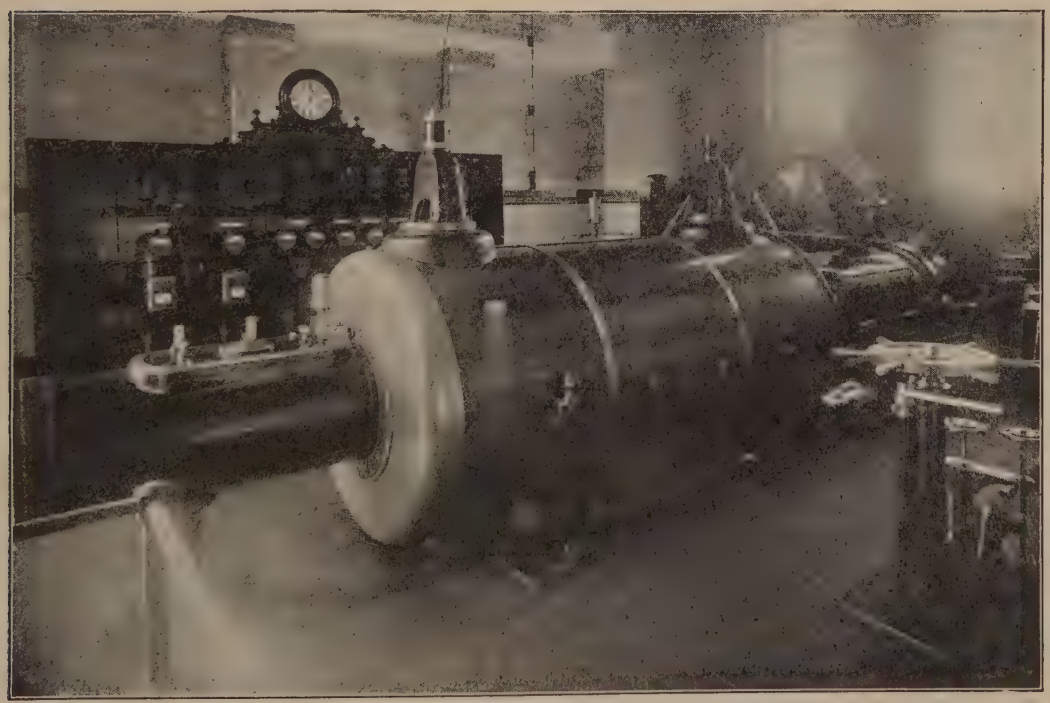


FIG. 58.—Another View of the Robey 500 b.h.p. Uniflow Engine, showing Cylinder and Valves. The Generator Switchboard is also shown.

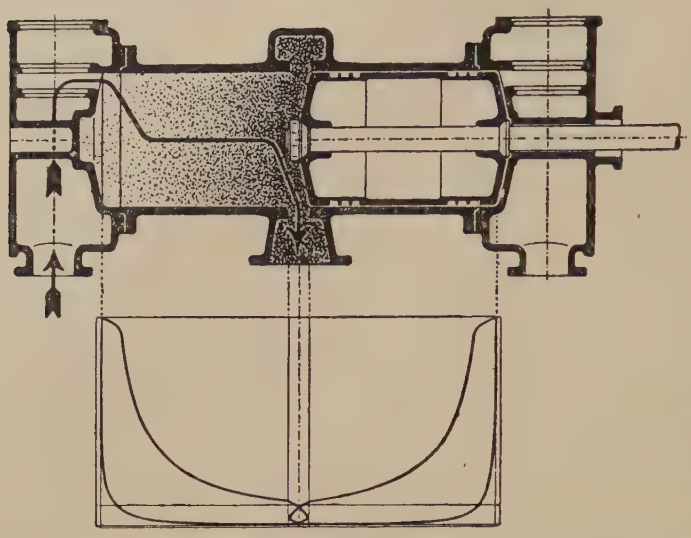


Fig. 59.—Diagrammatic Section through Cylinder of Galloway Uniflow Engine.

which exhausts direct to a condenser. The engine is coupled to a direct current generator having an output of 360 kw. at a pressure of 230 volts. The massive flywheel may be noted, this being fitted in order to give the necessary uniform turning moment when driving an electric generator from a single-crank engine. Tests carried out by the consulting engineer for the purchasers of the set showed that under the specified conditions the steam consumption was 10·63 lb. per i.h.p. per hour, which is a very efficient performance considering the comparatively small power of the unit and the low boiler pressure employed.

The uniflow engine is now being manufactured by a number of the leading engine builders in this country, and as a further example of this type and one showing a number of very interesting details, the engine designed and constructed by Messrs. Galloways, Ltd.,

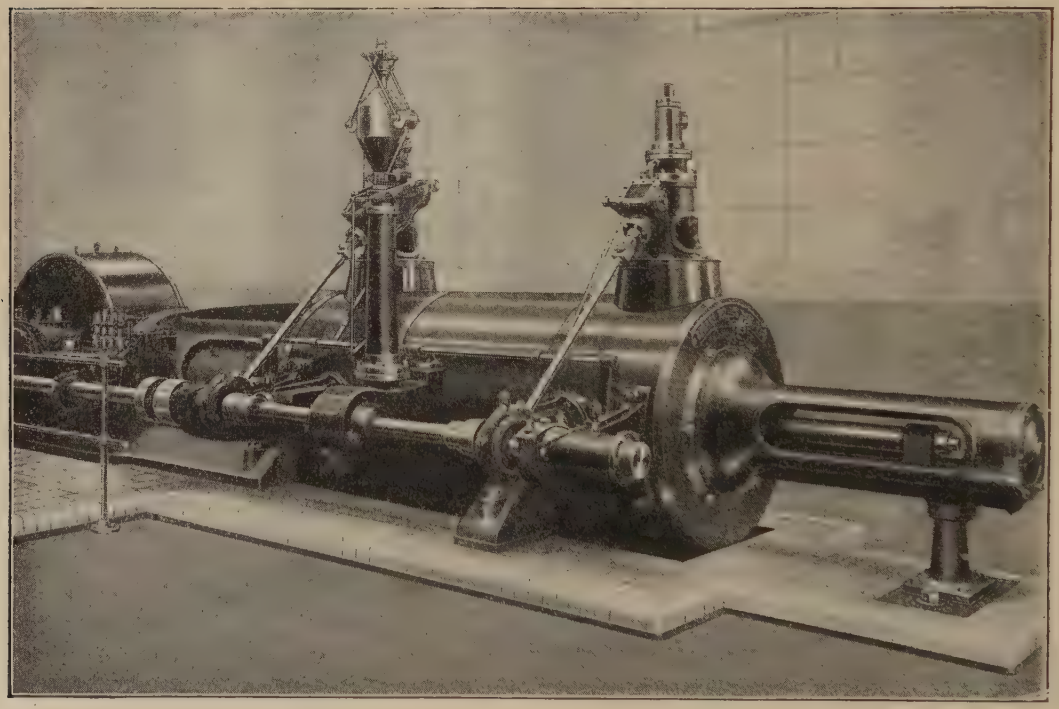


Fig. 60.—Single-cylinder Uniflow Engine by Messrs. Galloways, Manchester. View showing Admission Valve Gear and the Governor.

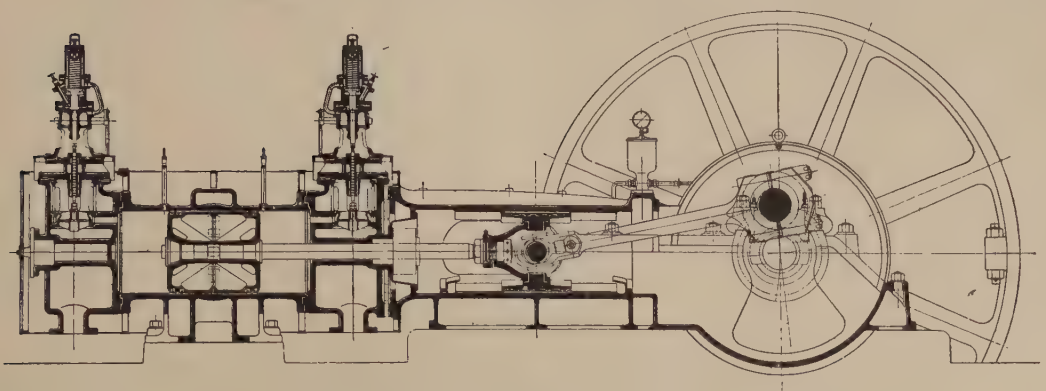


FIG. 61.—Sectional Elevation of Galloway Uniflow Engine, showing size of Piston.

of the Knott Mill Iron Works, Manchester, is worthy of study. The principle upon which the engine works is precisely the same as that adopted in the uniflow engine already described. Fig. 59 is a diagrammatic representation of a section through the cylinder of the Galloway uniflow engine, and below it are reproduced the indicator diagrams for both sides of the piston, each in its correct position relatively to the stroke of the piston. The arrows indicate the direction of the steam through the cylinder from the admission valve through the exhaust ports to the condenser, and the black dots are intended to represent the gradual increase in the percentage of moisture as the steam passes through the engine. These may also be taken as indicating the decrease in the temperature of the steam and with it that of the cylinder walls, the lowest temperature being, of course, at the exhaust ports.

The general appearance of one of these engines will be gathered from Fig. 60, which is a view showing the admission valve gear and the governor. It will be noticed that the crank on the extreme left and the crosshead are totally enclosed and that the piston is provided with a tail rod fitted with a crosshead working on a guide. This serves to take the weight of the piston and thus relieves the tendency towards uneven wear of the cylinder liner—a matter of considerable importance with large cylinders and where a piston of considerable weight, as in the uniflow engine, is used. In the photograph the flywheel is not shown, being just outside the picture, but with a single-cylinder engine—this being the arrangement generally adopted where the uniflow principle is employed—a heavy flywheel is absolutely essential in order to yield a uniform turning effort at the shaft. A sectional elevation of one of these engines is shown in Fig. 61, in which many of the distinctive features of the design are clearly shown. It will be noticed that the crank, which is of the double-web type, is provided with heavy balance weights in order to give smoothness of running. The connecting rod is provided with a bottom or big end of the marine type, the "brasses" being of steel lined with white metal, while laminated liners are fitted to

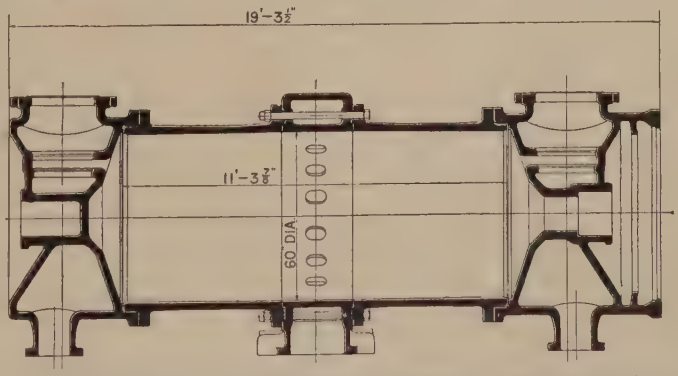


FIG. 62.—Section through Built-up Cylinder of Galloway Uniflow Engine of 2,700 i.h.p.

provide for adjustment. The crosshead end is fitted with phosphor-bronze bearings with wedge adjustment. The crosshead itself is of cast steel fitted with turned shoes which in the larger sizes and for high speeds are lined with white metal. The form of the piston is clearly shown, and it will be noticed that two sets of rings are fitted, one at each end. The rings are made of cast iron and machined by an improved method which, the makers claim, ensures a known uniform pressure all round the cylinder barrel, the rings being tested in a special machine to make certain that the desired pressure is obtained and that it acts uniformly at all parts of the ring.

A special feature of the Galloway uniflow engine is the cylinder, which in the largest engines is of a patent built-up pattern. Messrs. Galloways have constructed what is believed to be the largest uniflow engine in the world, one having a cylinder 60 in. in diameter

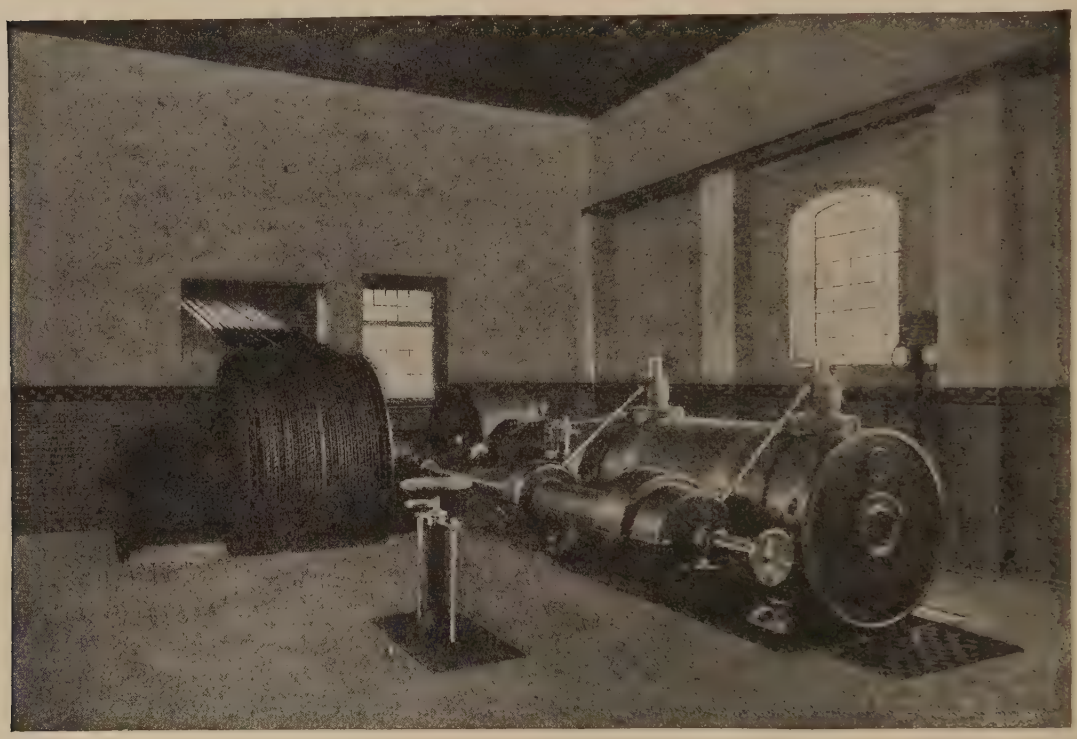


Fig. 63.—" Central Exhaust " Engine of 760 max. b.h.p., by Messrs. Cole, Marchent & Morley, Ltd. The Engine drives a Tinplate Mill.

with a piston stroke 72 in. A section through the cylinder of this engine is shown in Fig. 62, from which it will be noticed that the cylinder barrel is 11 ft. $3\frac{7}{8}$ in. long, while the overall length is 19 ft. $3\frac{1}{2}$ in. The cylinder is built up in five parts, the central portion carrying the exhaust ports, the cylinder barrel, which consists of two simple cylindrical castings, bolted one on each side of the centre portion, and the two end portions carrying the valves. From these latter it will be noticed that the valves are double seated and that the steam enters by a connection made under the cylinder, the valve gear being mounted on the top as shown in Fig. 61. This form of cylinder construction with its simplified barrel portions is found to be well suited, in view of its great length, to withstand the effects of highly superheated steam, which tends to distort all complicated castings due to unequal expansion in various directions. Another interesting feature—not shown on the drawing—is a patent inspection door and reflecting mirror

device, by means of which the internal condition of the cylinder wall and of the piston rings can be ascertained without dismantling the engine.

The steam admission valves are operated by means of Galloway's patent oil-valve gear, which has already been described and illustrated in dealing with the same firm's drop-valve engine (see page 43). The oil pressure which operates the valves is obtained by means of a small plunger driven by an eccentric mounted on the lay shaft of the engine, as shown in Fig. 60. This lay shaft also carries the gear which drives the governor, although when desired, instead of the vertical centrifugal governor, as shown, a governor mounted directly on the lay shaft can be employed if circumstances permit. The lubrication to all parts of the engine is by means of a small pump driven by an eccentric mounted on the lay shaft. The oil is drawn from a storage reservoir by the pump which delivers it under pressure to a small tank mounted on the engine frame,

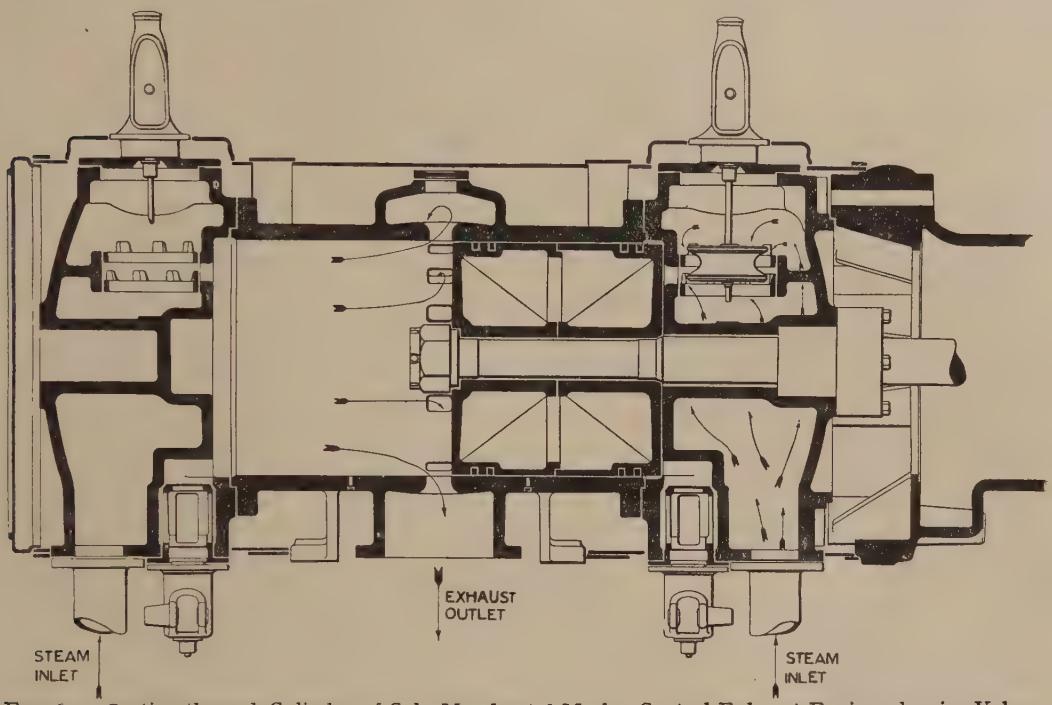


FIG. 64.—Section through Cylinder of Cole, Marchent & Morley Central Exhaust Engine, showing Valves and Exhaust Ports.

whence it is circulated through sight feed separately controlled connections to the main bearings, the crank-pin bearing, the gudgeon-pin bearing, and the crosshead shoes. A special mechanically operated sight feed lubricator is used for the cylinder, which is flushed at starting with a hand pump. This pump can be discerned in Fig. 60 just by the main lubricating pump.

Reference has already been made to the engine having a cylinder 60 in. diameter with a 72 in. piston stroke. This engine is capable of developing normally about 2,700 i.h.p. at a speed of 63 revs. per min. with superheated steam at a pressure of 160 lb. per sq. in. and a cut-off of about 10½ per cent. of the stroke, the mean effective pressure on the piston being 42 lb. per sq. in. The output of this engine is 2,460 b.h.p., but in view of the low mean effective pressure it will be easily understood that the engine possesses a large overload capacity. A very good idea of the flexibility of these engines will be gathered from the fact that one unit of the above dimensions was designed for a

normal running speed of 28 revs. per min., the engine being direct coupled to a sheet metal mill. At the other end of the scale the units have a cylinder 17 in. diameter with 21 in. piston stroke, the indicated horse power being 180 at a speed of 180 revs. per min.

Another interesting form of uniflow engine is that which is designed and constructed by Messrs. Cole, Marchent & Morley, Ltd., of Bradford, Yorkshire. This firm sometimes refer to these engines as their " central exhaust " type, a name which obviously conveys the idea that the exhaust steam passes through ports arranged at the centre of the cylinder. The general principle underlying the operation of the engine is the same as that for the other uniflow engines already described, but in the design a number

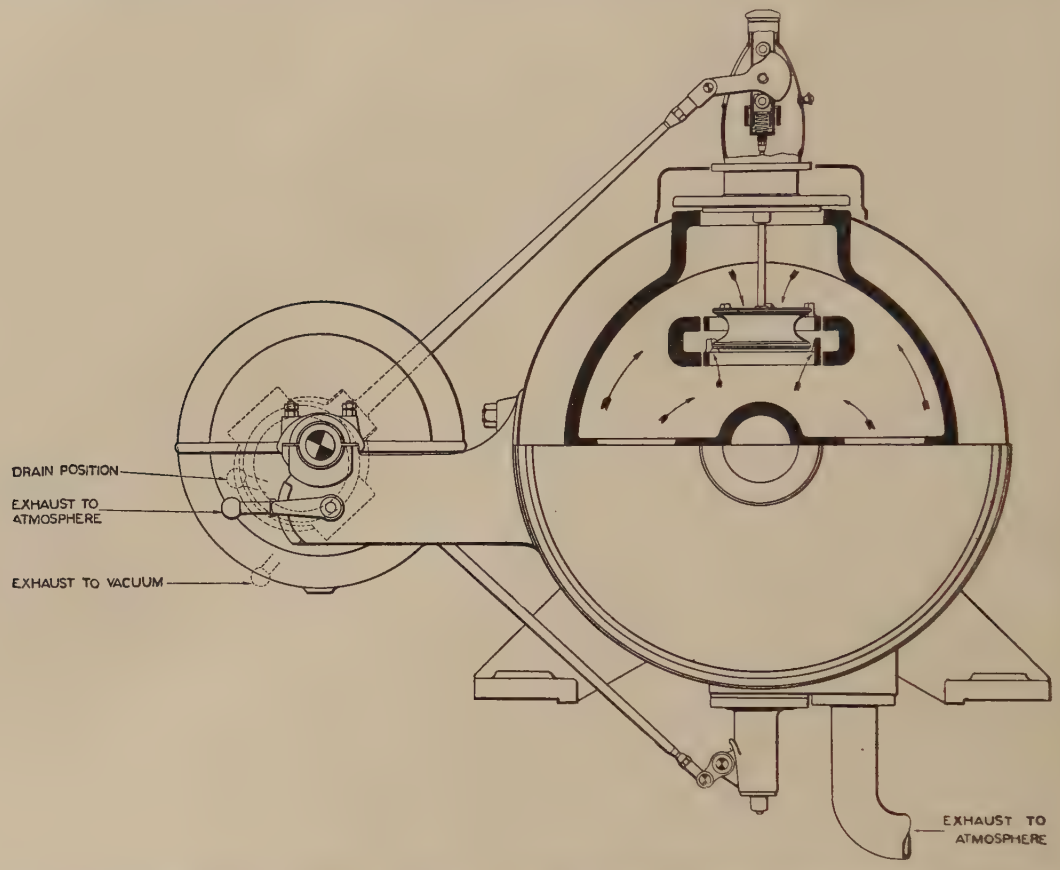


FIG. 65.—Sectional End Elevation of Cylinder of Cole, Marchent & Morley Engine, showing Valve Gear.

of unique features have been introduced. The general appearance of these engines will be gathered from Fig. 63, which shows a unit installed for driving the machinery in a tinplate works in South Wales. The engine has a heavy flywheel which is grooved for rope driving, the mill being situated in another building, a line of railway track separating the two. The top ropes pass over the roadway and the return ropes pass through a culvert beneath the wagon road. It may be noted in passing that the mill shaft is 24 in. diameter and is carried in bearings 54 in. long, while the mill wheel is 32 ft. in diameter and weighs approximately 120 tons. It is driven at a speed of 36 revs. per min. Returning to the engine, it will be seen that this has a single cylinder with lay shaft mounted at the side, the valve-operating gear and the governor, all of which are mounted on the lay shaft being totally enclosed, the governor being of the centrifugal

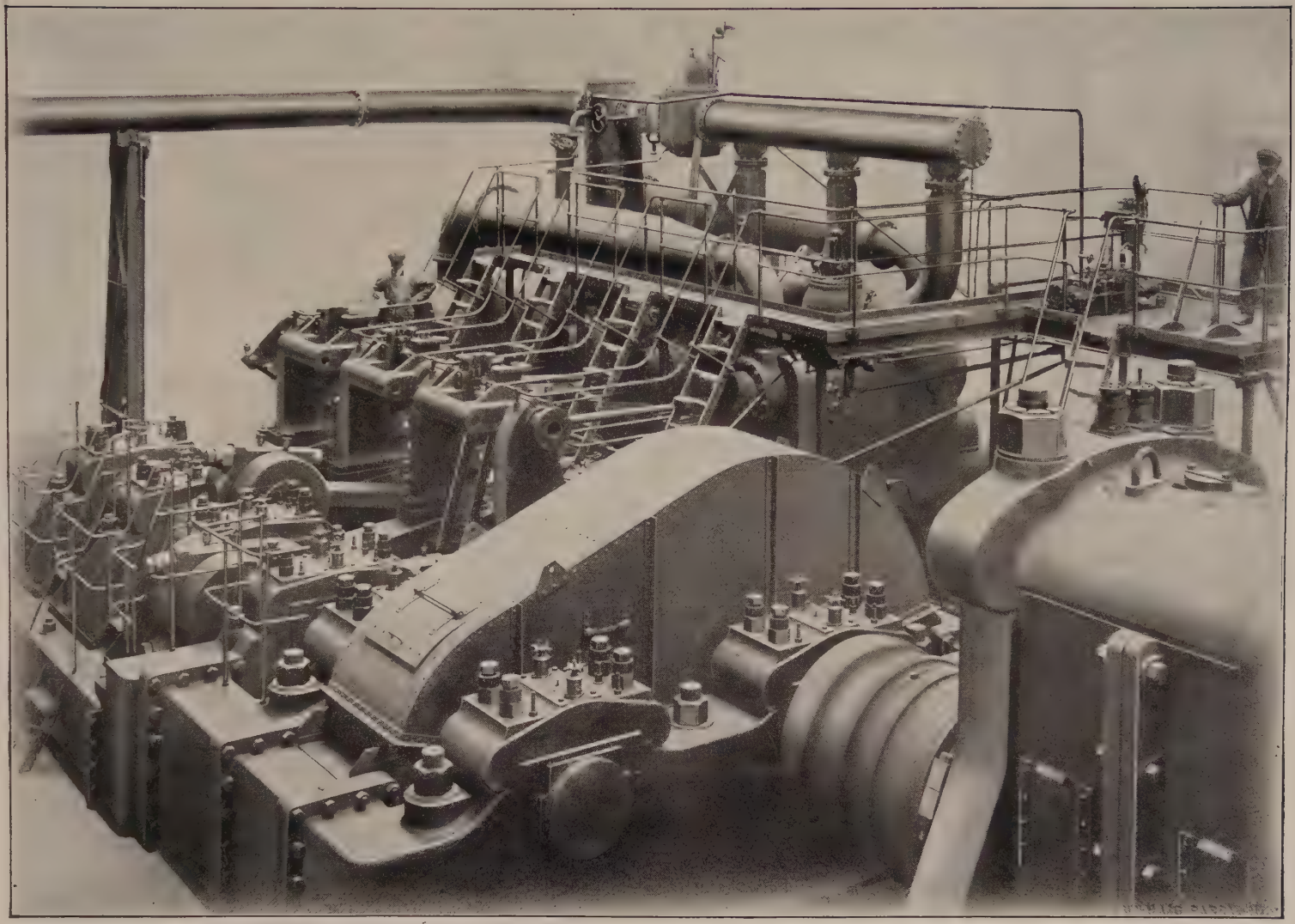


PLATE II.—THREE-CRANK GEARED REVERSING ROLLING MILL ENGINE BY MESSRS. GALLOWAYS, LTD., MANCHESTER.

type with weights rotating about the lay shaft as an axis. The cylinder is 30 in. diameter and the stroke of the piston is also 30 in. Steam is supplied at a pressure of 130 lb. per sq. in. superheated to a temperature of 575° F., and the designed speed of the engine is 144 revs. per min. The normal rating is 600 b.h.p., while the maximum continuous capacity is 760 b.h.p. The flywheel is 8 ft. diameter and weighs approximately 18 tons, while the rim is grooved to take 20 ropes each 2 in. diameter.

A section through the cylinder of one of these engines is shown in Fig. 64, from which it will be noticed that this is in three parts, the cylinder barrel and the two covers, which are of similar design but arranged in opposite directions. The piston is built up in two pieces bolted together as shown and is provided with piston rings at each end to prevent leakage of steam from one side to the other. The valves, which are of the double-ported type, work in cast-iron liners which are pressed into position in the cylinder cover. The valves are operated by a patent gear having a positive motion both for opening and shutting. The general arrangement of the valve gear is shown in Fig. 65, which is an end elevation of the cylinder, the top half being in section taken through the steam chest, while Fig. 66 shows the valve and the operating gear as assembled before mounting on the engine. It will be noticed that the valve gear is operated by means of an eccentric mounted on the lay shaft, while leading downward from the lay shaft to the underside of the cylinder is another rod which might at first sight be taken for a rod operating the exhaust valve. As a matter of fact, this is the operating gear for a combined drain, exhaust, and relief valve, there being one of these at each end of the cylinder as shown in Fig. 64. The operation of this gear will be easily understood from Fig. 67, which shows diagrammatically the driving cam and the control lever. This control lever has three working positions, the lowest of which, i.e. that marked "exhaust to vacuum," is the position when the engine is running under normal conditions with the exhaust steam passing to the condenser through the ports at the centre of the cylinder. When the lever is in this position, the roller is clear of the cam and the valve in the cylinder remains closed, despite the rotation of the cam.

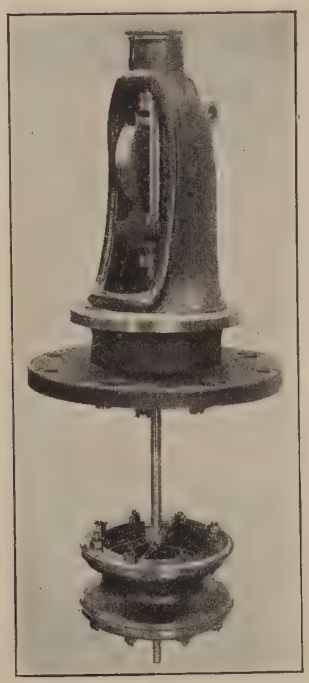


Fig. 66.—Admission Valve and Operating Gear before Bolting to Cylinder.

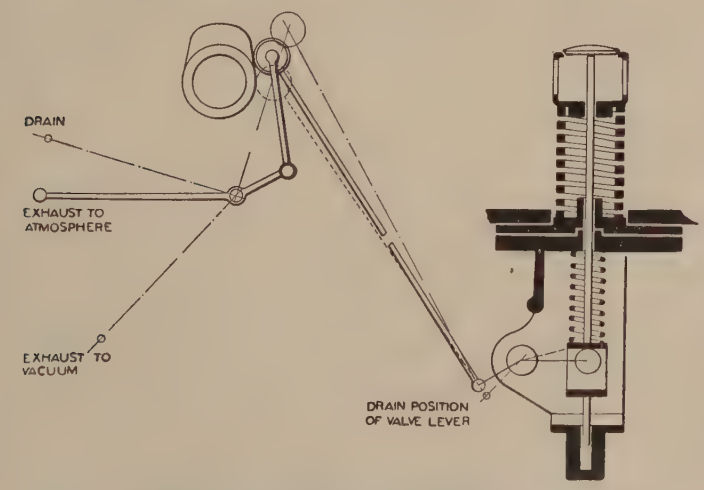


Fig. 67.—Combined Drain, Exhaust, and Relief Valve and Gear for Cole, Marchent & Morley Central Exhaust Engine.

On starting up the engine it is desirable that the steam should be permitted to blow through the cylinder, carrying with it

any water that may have accumulated in the cylinder, although it is undesirable that this water should find its way into the condenser, owing to the oil which may be mixed with it, which might ultimately find its way into the boiler. Hence, means are provided for enabling the steam to blow through the cylinder—generally into the atmosphere, or if the presence of oil is not feared, into the condenser. When the control lever is in the highest position, i.e. that marked "drain," the valve is opened, and the position of the roller is such that it remains open even when the engine is running. It may be desirable when the boiler shows a tendency to prime, i.e. when a certain proportion of moisture finds its way over with the steam, a condition which frequently arises, particularly if the steam is not superheated, to open the drains occasionally, even while the engine is working on load. This gear enables such an operation to be easily effected. Finally, if the condenser is for any reason placed out of service, it may still be necessary to have the engine running. Under these circumstances the engine may be run non-condensing, the exhaust steam passing to the atmosphere. When the control lever is placed in the intermediate position which is marked "exhaust to atmosphere," the roller is brought down on to the cam and the valve is operated at the correct time to allow the steam to be exhausted on the return stroke of the piston, thus enabling the engine to be run as a non-condensing drop valve engine. It must be remembered that under these circumstances the power developed by the engine will be somewhat reduced and the efficiency will be decreased; but if it is possible by the use of this gear to keep the engine in operation despite a breakdown of the condensing plant, it may prove of very great value to have such gear fitted. In order to facilitate starting up, a small barring engine is provided, the form of which is shown in Fig. 68. This engine drives a worm which drives a worm wheel, and on the shaft carrying the worm wheel is a toothed wheel which gears with a larger wheel and thence with the teeth which are provided on the rim of the flywheel of the main engine. The larger-tooth wheel is carried on a throw-out lever, so that the starting gear can be engaged and disengaged when desired, although, as the main engine gains speed, the gear wheel is automatically thrown out, after which the starting engine can be shut down.

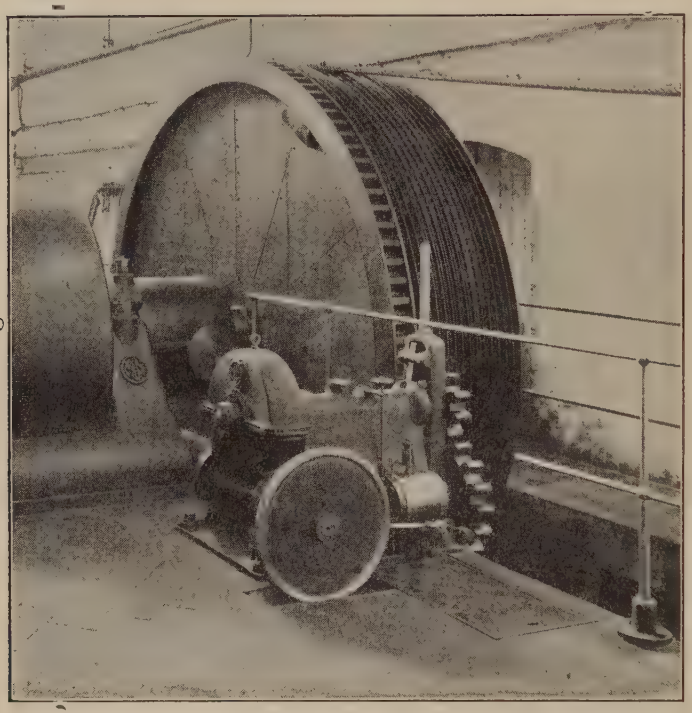


FIG. 68.—Barring Engine for large Cole, Marchent & Morley Central Exhaust Engine.

In addition to the single-cylinder condensing engine which Messrs. Cole, Marchent & Morley make in various sizes between 100 and 3,000 i.h.p., this firm have produced

special types of uniflow engine adapted for heat extraction purposes for use in factories where steam is required at low pressure for heating or for other industrial purpose. If only a portion of the steam used by the engine is required for heating, a compound engine is recommended in which the steam after doing work in the high-pressure cylinder is passed to a receiver from which the heating steam is extracted at constant pressure, while the remainder is passed to the low-pressure cylinder where it expands further, doing useful work before passing to the condenser. Should the whole of the steam used by the engine be required for heating purposes, a single-cylinder non-condensing engine is employed which exhausts at a pressure slightly above atmospheric into the heating system. Fig. 69 shows a heat extraction engine of the former type which is installed in a paper mill in the south of England. As will be seen, the engine is of

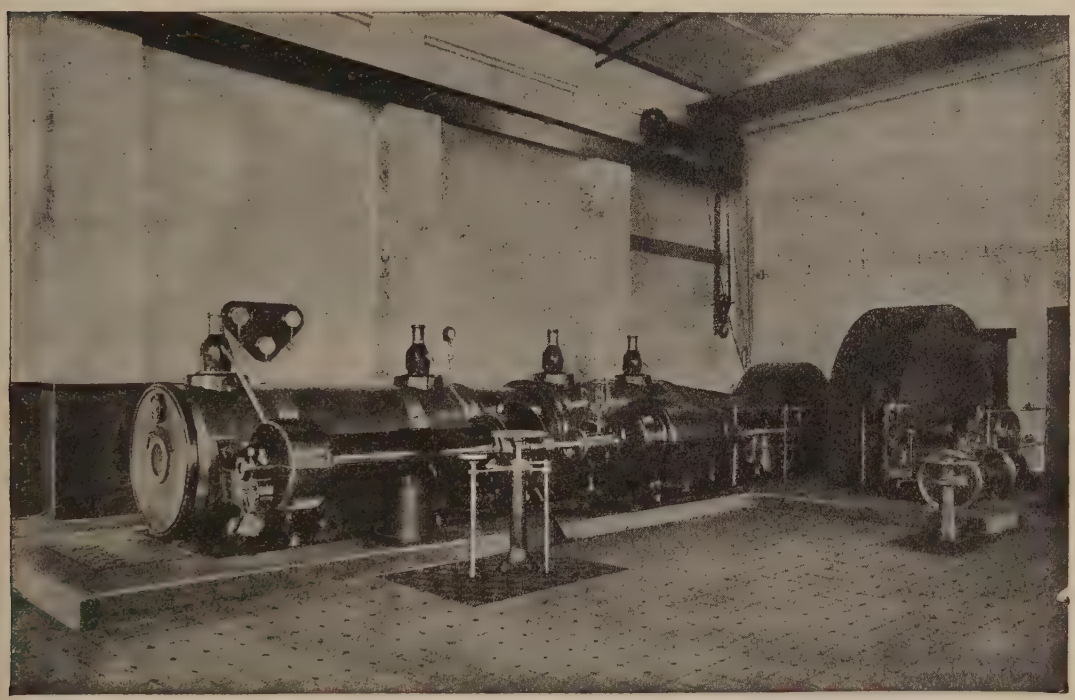


FIG. 69.—Tandem Compound Extraction Engine by Messrs. Cole, Marchent & Morley, Ltd., passing 8,000 lb. of steam per hour at 10 lb. per sq. in. for Heating Purposes.

the tandem compound type, having a high-pressure cylinder of the usual four-valve pattern with mechanically operated drop valves for both admission and exhaust, and a low-pressure cylinder of the central exhaust pattern. Between the two there is a receiver from which steam is extracted at constant pressure, a patent automatically regulated device being employed to ensure a constant pressure in the heating system. The surplus steam passes to the low-pressure cylinder. At full load the engine is designed to give 8,000 lb. of steam per hour from the receiver at a pressure of 10 lb. per sq. in., while using boiler steam at a pressure of 160 lb. per sq. in. superheated to a temperature of 500° F. the power developed by the engine is 530 i.h.p. at a speed of 150 revs. per min.

As an example of an engine in which the whole of the exhaust is required for heating, the unit shown in Fig. 70 may be considered. This engine, which is also installed at a cardboard and paper mill, is of the single-cylinder central exhaust type. Steam is supplied at a pressure of 150 lb. per sq. in. superheated to a temperature of 400° F., and

the engine exhausts against a back pressure of 15 lb. per sq. in. The normal power developed by the engine is 1,260 i.h.p. at a speed of 140 revs. per min.

One of the most arduous duties which an engine can be called upon to perform is the driving of steel rolling mills. In the first place, the load fluctuates between wide limits, changing suddenly from that required merely for operating the rolls when running light to that required when the steel billet is being passed through the rolls. Further, owing to the exigencies of furnace operation, the rolling mills are usually run continuously night and day for five days per week or longer, which means at least 120 hours' continuous operation. It has been found, as a result of long experience, that the reciprocating steam engine is the best type of prime mover which has yet been devised for the

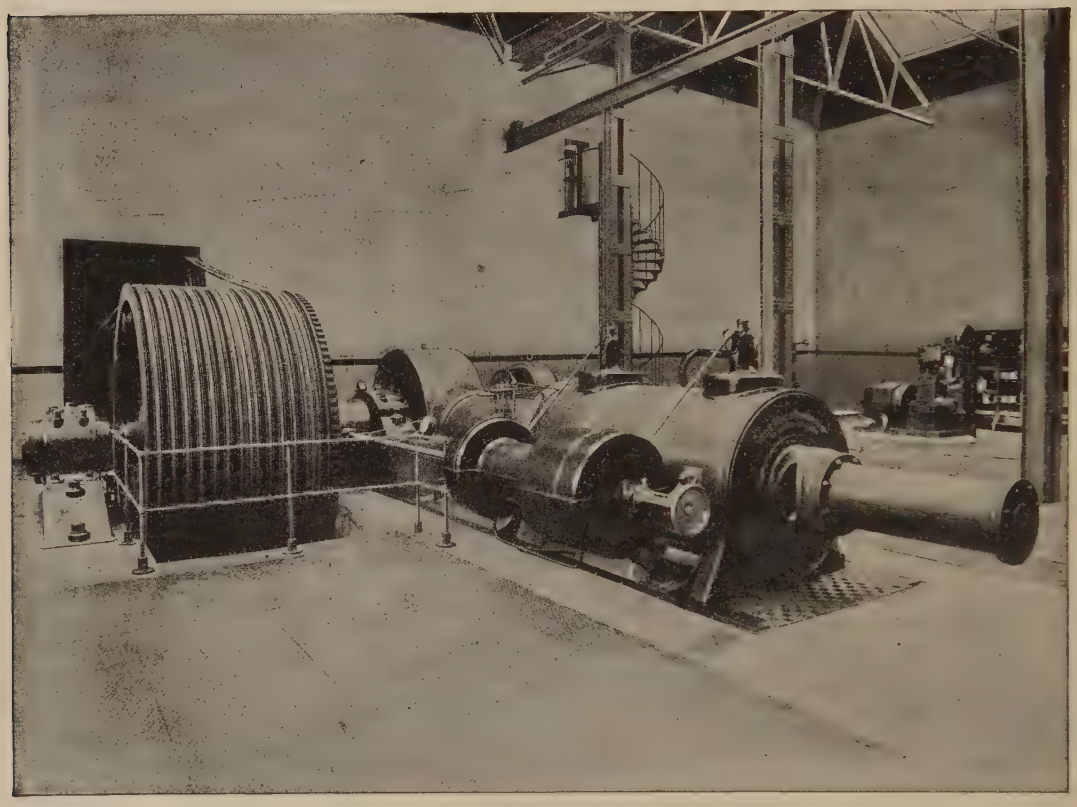


FIG. 70.—Cole, Marchent & Morley Back Pressure Central Exhaust Engine of 1,260 i.h.p., exhausting at a pressure of 15 lb. per sq. in. above atmospheric.

duty of driving rolling mills. Two forms of engine are available: (*a*) the continuous running engine, in which a mechanism is employed between the engine and the rolls to obtain a reversing motion at the latter, thus enabling the billet to be passed backward and forward through the rolls; and (*b*) the reversing engine, in which the rolls are driven direct from the engine, reversal of the rolls being obtained by reversing the direction of running of the engine. When large powers are involved, the provision of gearing for reversing the direction of the rolls becomes a matter of very great difficulty, and in any case such gearing is always a source of weakness; hence the reversing type of engine is generally employed.

As an example of a reversing rolling mill engine we can consider the engine shown in Plate II, which is a reproduction from a photograph of a large three-cylinder three-

crank engine designed and constructed by Messrs. Galloways, Ltd., Manchester. The man standing on the control platform serves to give some idea of the vast size of the engines of this type. The general arrangement drawing of this engine is reproduced in

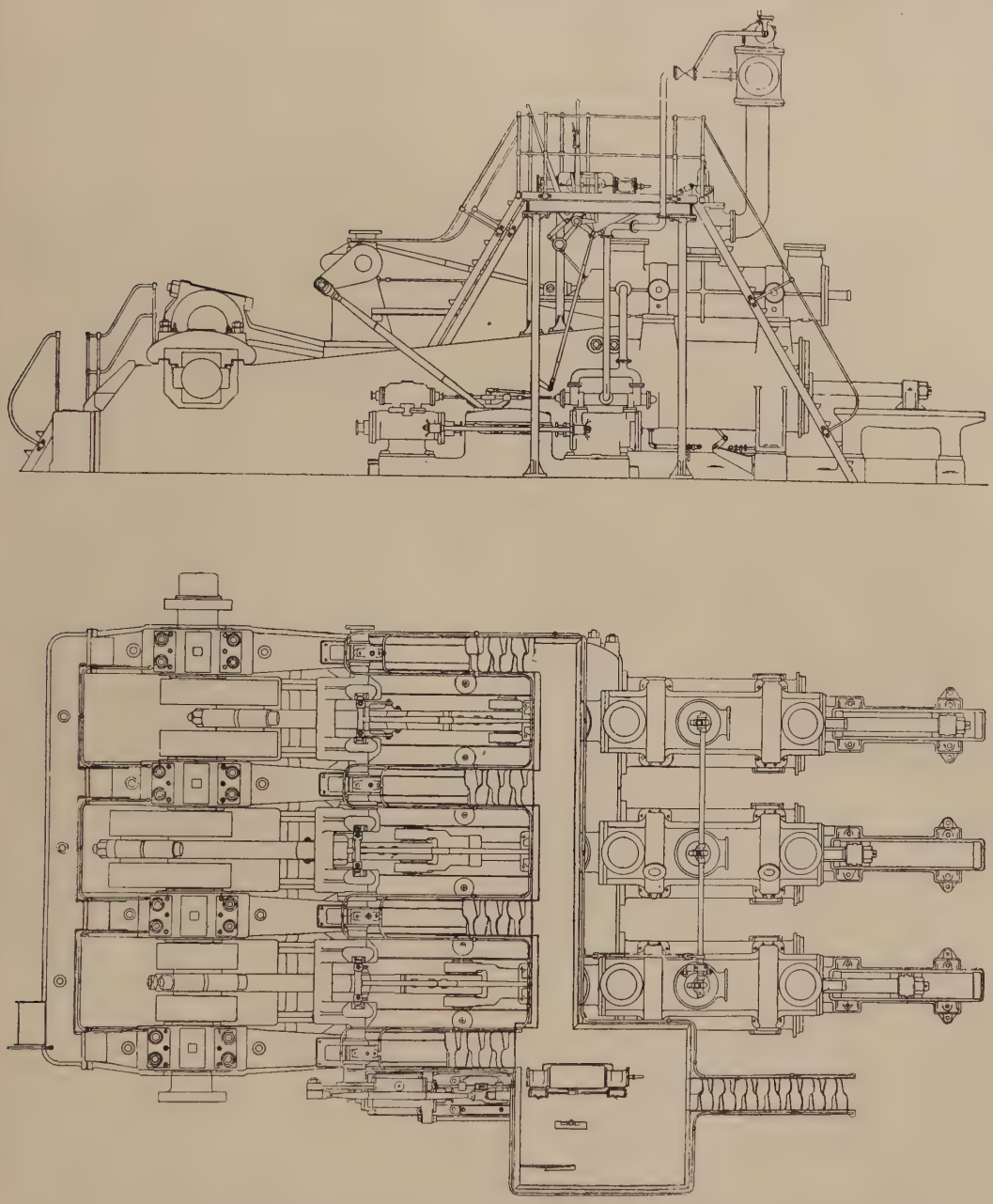


FIG. 71.—General Arrangement Drawings of Three crank Simple Reversing Rolling Mill Engine by Messrs. Galloways, Ltd., Manchester.

Fig. 71, from which it will be seen that the three cylinders are all of the same diameter, each receiving steam direct from the boiler stop valve through a junction pipe which is clearly seen in Plate II. Steam is distributed to the cylinders by means of ordinary piston slide valves working in valve chests mounted directly over the cylinders, and

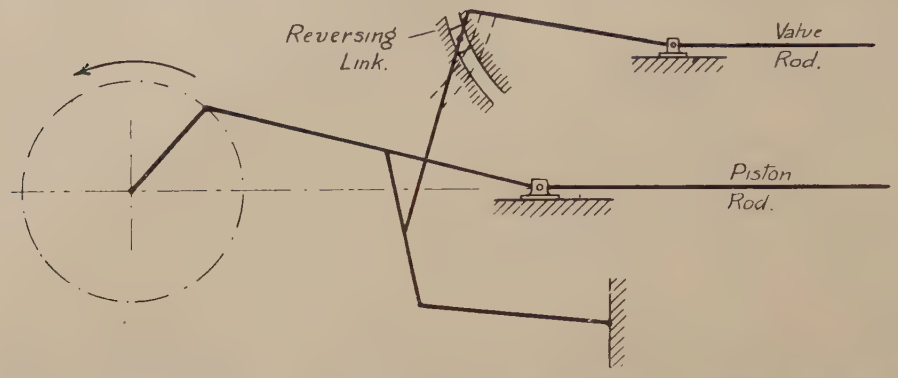


FIG. 72.—Outline Diagram of Reversing Steam Engine, showing Arrangement of Joy's Valve Gear.

after expansion in the cylinders it passes directly to the condenser arranged in a pit at the rear of the engine. In the engine shown in Plate II the cylinders are each 40 in. diameter with a piston stroke of 4 ft. 6 in. A feature of these engines is the patent single-lever expansion control gear, by means of which the point of cut-off of the steam can be varied over a wide range between zero, i.e. at the latest possible moment during the piston stroke and very early in the stroke, thus affording the maximum economy for very widely fluctuating loads. The piston valves are driven through expansion gear of the " Joy " pattern, by means of which eccentrics or cams are eliminated, the valves being operated through a series of levers driven from the connecting rod of the engine itself. The arrangement of Joy's valve gear is shown diagrammatically in Fig. 72. With this gear, reversing is effected by moving the link from the position shown in full to the position shown dotted, the extreme positions yielding the latest possible cut-off, or in other words the maximum power developed by the engine. For intermediate positions of the link the cut-off becomes earlier and earlier in the stroke, until when the link is in mid-position there is no distribution of steam to the cylinder, i.e. the engine is stopped. Some idea of the wide range of expansion which is possible can be gathered from Fig. 73, which shows a series of indicator diagrams taken from one of the cylinders of a three-cylinder reversing mill engine fitted with Joy's valve gear. It will be noticed that the mean effective pressure on the piston is varied from 84·8 lb. per sq. in. at full load down to 4·2 lb. per sq. in.

Control is effected by means of a single lever mounted on the driving platform, which is coupled through two systems of links, one to the small piston valve of the steam reversing servo-motor which is mounted at the side of the engine (see Fig. 71), and the other to the steam-handling valves, one of which is mounted on each cylinder. These valves are of the double-beat type, and when the control lever is in its mid-position they are shut so that no steam can enter the cylinders. At starting, the control lever is moved either to the forward or backward running position, the steam-handling valves are slightly opened, and the link is moved to the position in which a very early cut-off occurs, with the result that the engine just creeps slowly round. When, however, the

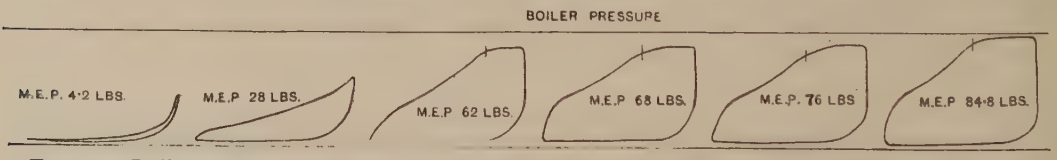


FIG. 73.—Indicator Diagrams taken from a Galloway Rolling Mill Engine, showing Range of Power obtained with the Patent Expansion Control Gear.

rolls grip the billet or ingot of metal, the driver thrusts the lever forward into full gear. The valves are then fully opened and the latest cut-off is obtained, the steam being at full pressure, thus giving the maximum torque and causing the engine to accelerate immediately. During the rolling period any linking up will give a partial throttling at early cut-off, but when the material is about to leave the rolls, the driver's lever is brought into the central position again, thus bringing the valve motion into mid-gear and closing the handling valves. Thus the engine is quickly brought to rest and is ready for the next pass of the material in the opposite direction. It is claimed that with this control gear, despite the fact that the intermittent running of the engine is inimical to

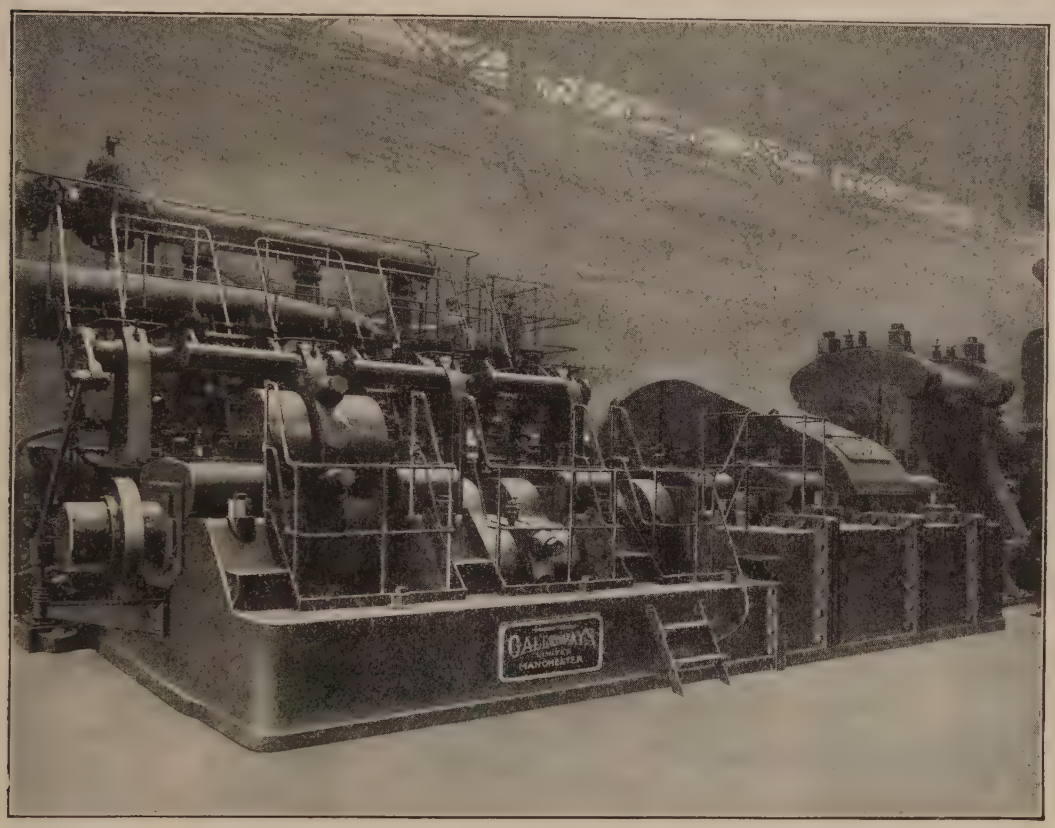


FIG. 74.—Front View of Galloway Reversing Rolling Mill Engine, showing Crank Shaft and Bed Plate.

economical performance, the maximum output is obtained with the lowest possible steam consumption.

In view of the fluctuating load on these engines, very rigid construction is called for both in the frame and in the working parts. The bed frame is therefore of very massive and compact construction, being of the built-up type with the longitudinal members of deep box section continuous from the cylinder end to beyond the main bearings. The main bearing ends of the longitudinal members and an intermediate point near the centre of the slides are connected by rigid cross stringers. The various members are joined together by internal flanged connections secured by mild-steel bolts, a suitable number being "fitted" into reamered holes. The bed frame is supported along the whole of its length on the concrete foundation to which it is securely anchored by bolts. The main crank shaft is supported on four massive bearings, as shown in Fig. 74, which is a view taken from the front of the engine, these bearings being turned

and finished in one operation to ensure perfect alignment. The engine drives the rolling mill through suitable machine-cut gearing which works in an oil bath and gives the necessary speed reduction between the engine and the rolls.

For modern high-powered, slow-speed, horizontal, reciprocating steam engines the drop valve or the uniflow principle is almost invariably employed. There still remains, however, another type of engine to which reference must be made, and that is the engine in which the well-known " Corliss " valve and valve gear is employed. It must be admitted that this type of engine is not being built in any large numbers at the present day, but there are still very many " Corliss " engines to be found performing all kinds of duties in industrial establishments all over the world, and as it is our desire to make

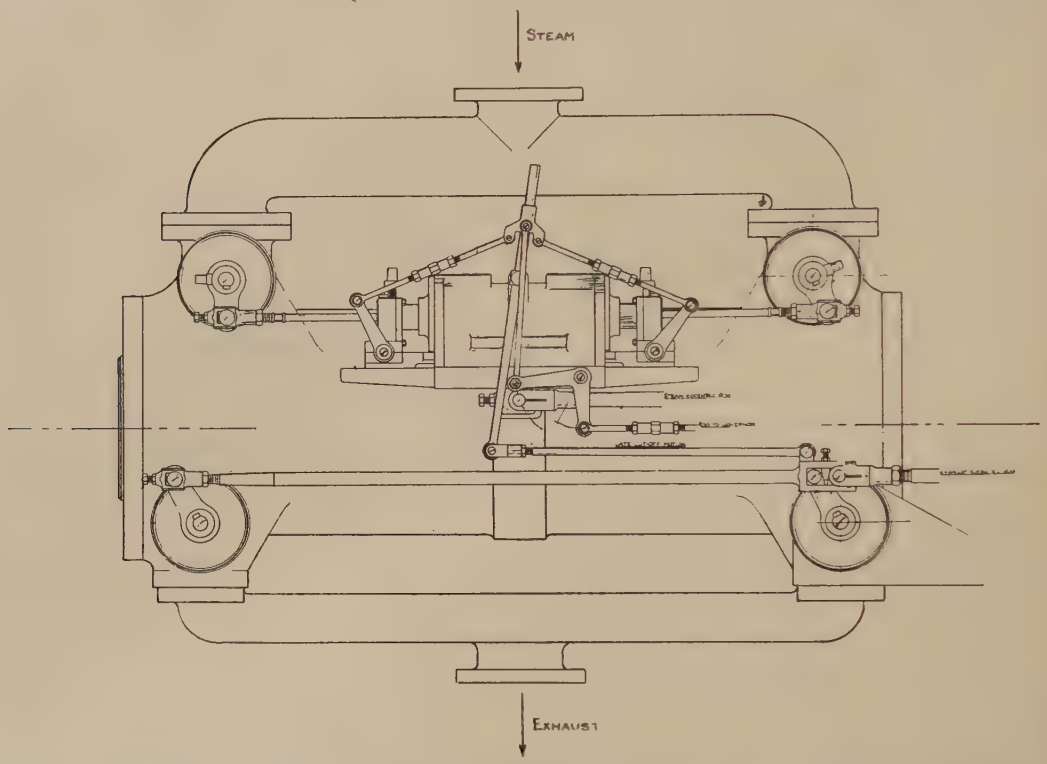


FIG. 75.—Elevation of Cylinder of " Corliss " Engine, showing Admission and Exhaust Valves and Trip Gear operating the Admission Valves.

these volumes as comprehensive as possible and to describe not only the most modern forms of power generators but also those which are still to be found in very large numbers, although perhaps gradually becoming obsolescent, a description of the " Corliss " engine must be given. Before, however, we deal with the engine itself, it will be of interest to give a brief description of the " Corliss " valve arrangement and the method whereby the valves are operated.

The " Corliss " valve and valve gear received its name from an American engineer, Mr. George H. Corliss, who, about the year 1850, recognising the disadvantage of having a single valve to control both the admission of the steam to and the exhaust of the steam from the cylinder, devised an arrangement of separate admission and exhaust valves together with the gear for operating them, to which, despite many subsequent modifications and improvements, his name has been given. The general arrangement of the valves on the cylinder of a " Corliss " engine will be understood from a study of

Fig. 75. It will be noticed that the valves are of cylindrical form, with their axes at right angles to the centre line of the cylinder. They have an oscillating motion about their axes which in the early designs was obtained from a wrist plate mounted somewhere near the centre of the cylinder, the wrist plate receiving an oscillating motion from a single eccentric keyed to the crank shaft. The admission valves are at the top of the cylinder with the exhaust valves at the bottom, but whereas the latter are operated directly by rods from the wrist plate, some form of trip gear is used in conjunction with the former, this being under the control of the governor. In this way not only can the power of the engine be varied with the load by varying the point of cut-off, but a sharp cut-off is obtained which is conducive to high efficiency. Many different forms of gear have been devised to operate "Corliss" valves, and one of the best known of these is that patented by Messrs. Scott & Hodgson, Ltd., of the Guide Bridge Ironworks, near Manchester, and incorporated in the "Corliss" valve engines designed and manufactured by them. The general arrangement of this gear is

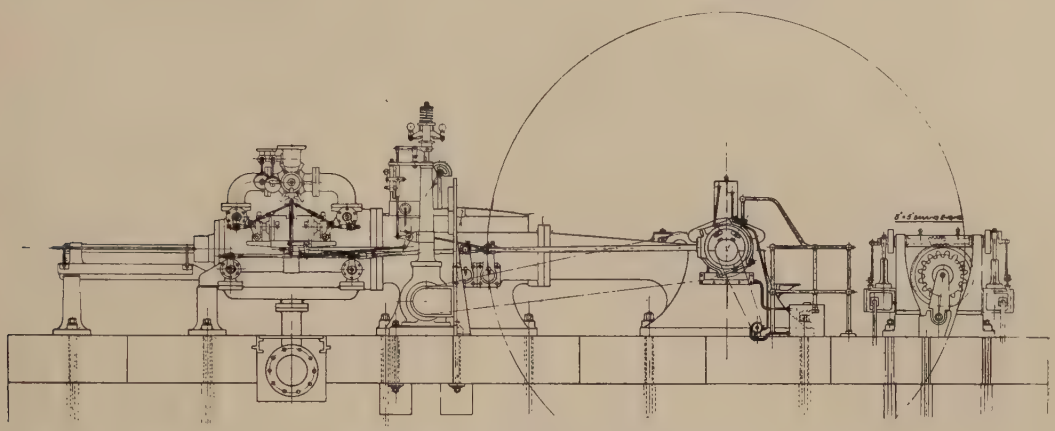


Fig. 76.—Elevation of High-pressure Side of a Cross Compound "Corliss" Engine of 400 i.h.p. by Messrs. Scott & Hodgson, Ltd.

also shown in Fig. 75, which is a side elevation of the cylinder of one of the engines designed by this firm. The position of valves at each corner of the cylinder is clearly shown, and it will be noticed that steam is supplied through a pipe having a flange at its centre upon which the engine stop valve is mounted. In a similar way the exhaust ports are connected by a pipe from which connection is made with the condenser. In this gear separate eccentrics are employed for operating the admission valves and the exhaust valves. The admission or steam eccentric rod, as it is called, imparts a reciprocating motion to a sliding block which, instead of driving the admission valves direct, operates the valves through a spring and a trip motion. The movement of the sliding block compresses a spring, which then moves the valve rod, causing the valve to open, admitting steam to the cylinder. At the required instant a cam held in position by a cam lever releases the spring, which immediately shuts the valve. The position of this cam can be regulated within wide limits and it is controlled by the governor in such a way that an earlier cut-off is obtained when the speed of the engine tends to increase and a later cut-off as the load increases tending to reduce the speed. This gear is found to be extremely reliable, giving a range of cut-off for all points of the stroke between the dead centre (i.e. practically no load) and seven-eighths stroke, and since the springs are compressed before admission of the steam takes place, no matter how early the cut-off may be the valve cannot stick. In this way a rather common trouble with "Corliss" valves is avoided.

Some idea of the general arrangement of a complete engine with " Corliss " gear will be obtained from Fig. 76, which is an elevation of the high-pressure side of a horizontal cross compound engine as erected in a cotton mill, the elevation being taken with the low-pressure cylinder removed. The engine is fitted with " Corliss " valves to both cylinders, but whereas the valves on the high-pressure cylinder are operated by the makers' patent trip gear as described above, the low-pressure cylinder valves are operated from a single wrist plate. The high-pressure cylinder is 15 in. diameter and the low-pressure cylinder 32 in. diameter, the stroke of the piston in each being 3 ft. 0 in. Steam is supplied from boilers having a working pressure of 160 lb. per sq. in., and at a normal running speed of 90 revs. per min. the engine develops 400 i.h.p. The flywheel, carried between the two cylinders, the centres of which are 9 ft. 6 in. apart, is 16 ft. in diameter and the rim is grooved to take twelve 1½ in. ropes, the inner periphery of the rim carrying

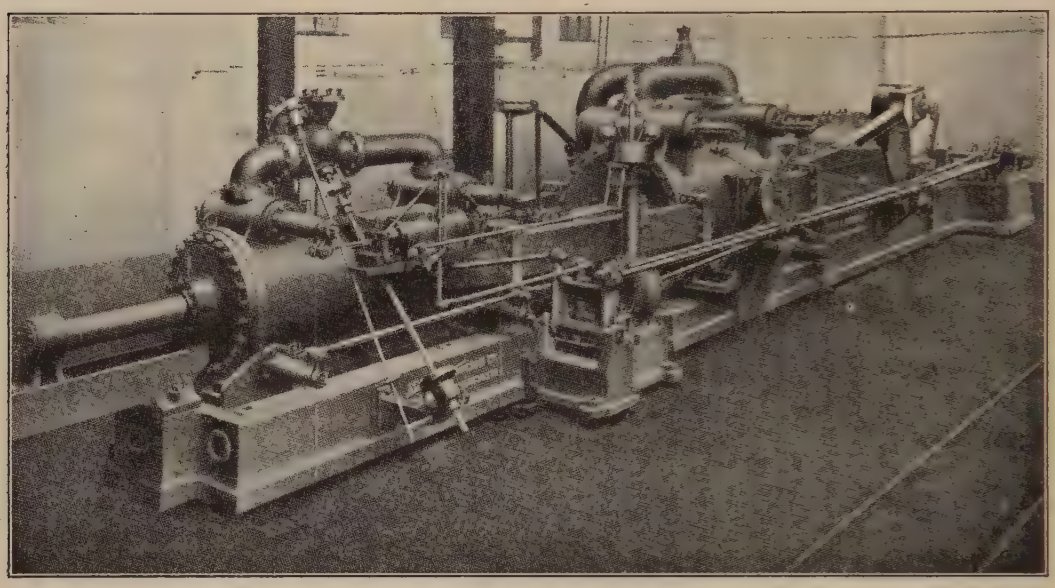


FIG. 77.—Tandem Compound Rolling Mill Engine of 2,500 i.h.p. with " Corliss " Valve Gear erected in the shops of the builders, Messrs. Scott & Hodgson, Ltd.

an internal toothed wheel which is engaged by a small toothed pinion driven by a small barring engine having twin cylinders 5 in. diameter with piston stroke 5 in.

A " Corliss " engine of larger size, also designed and constructed by Messrs. Scott & Hodgson, is shown in Fig. 77. This is a tandem compound engine developing a maximum of 2,500 i.h.p., designed for direct coupling to the roll shaft of a rolling mill, thus dispensing with the usual transmission gearing. The high-pressure cylinder is 35 in. diameter and the low-pressure cylinder 60 in. diameter, whilst the pistons have a stroke of 72 in. The flywheel for this engine is 34 ft. in diameter and weighs no less than 140 tons. The engine is seen in the illustration in course of erection in the makers' works and the flywheel is not in position. It will be observed on studying this illustration that the valves to the high-pressure cylinder are operated by the makers' patent trip gear, while a simple wrist plate is employed for operating the low-pressure valves. A very good idea of the general features of the engine, as well as of the peculiar form and arrangement of the " Corliss " valves, can be obtained from this illustration.

CHAPTER IV

PUMPING ENGINES—RECIPROCATING AND ROTARY PUMPS —THE HUMPHREY GAS PUMP

ONE of the most important applications of the reciprocating engine is in the driving of pumps. A pump can be defined as a piece of machinery which is employed to raise a liquid—usually water—from a lower level to a higher level, although very frequently it is not so much the level of the water that is changed as its pressure. Thus in an earlier chapter of this work we had occasion to deal with the auxiliary machinery which is to be found in every engine room where large steam power generating units are employed.[1] The majority of these auxiliaries consisted of pumps, but with the exception of the condenser circulating pump, all these were concerned rather with the creation of a difference of pressure than with raising water from one level to another. Thus, the boiler feed pump takes the water from the hot well, where it is generally at atmospheric pressure, and delivers it into the boiler, where the pressure is, say, 200 lb. per sq. in., so that the pressure of the water as delivered from the pump must be slightly in excess of this to enable the water to overcome the friction both in the pipe connecting the pump to the boiler and in the valve which controls the amount of water actually delivered to the boiler. With the condenser circulating pump, however, the pressure of the water when it enters the pump, and also when it enters the condenser, is approximately atmospheric, so that here the function of the pump is apparently quite different from that of the feed pump, being in the former case merely to raise water from a stream or other source of supply and deliver it through the condenser. In both cases the reciprocating type of pump, i.e. a pump consisting of a cylinder in which a piston—or pump bucket—moves backward and forward, can be conveniently employed, so that it will be obvious that the principle upon which both pumps are operated is more or less the same.

While, however, the pumps used in connection with the auxiliary machinery of a steam engine plant are of considerable importance, they are, comparatively speaking, of small size and require but small powers to operate them. It is when we come to consider the pumps which are employed in connection with the vast problem of providing an adequate supply of water to our large towns, or with the almost equally important problem of providing an efficient system of drainage from the small areas occupied by large masses of the population, that we realise how important a branch of engineering is the design and construction of pumps and what enormous capacities some of our modern pumps can have. In this connection attention may be drawn to Plate III, which shows a unit recently supplied to one of the pumping stations of the Metropolitan Water Board—the authority which is responsible for the supply of water to the whole of London—consisting of a horizontal compound engine driving double-acting pumps capable of delivering 4,500,000 gallons of water per day. We shall return to this unit at a later stage in this chapter, describing it in greater detail, but we have drawn

[1] See Volume I, Chapter VIII.

attention to it here as indicating the size of unit which is employed in dealing with the water supply to our large towns.

Statistics are notoriously uninteresting, but we may be pardoned if we present very briefly a few dealing with the supply of water to London, the largest city in the world. The area supplied by the Metropolitan Water Board, that is, Greater London, supports a population of over 7,000,000 persons, occupying an area of 514 square miles, and it is found that the average requirement of water for all purposes works out at about 32 gallons per person per day, so that the amount of water which has to be provided is no less than 224,000,000 gallons per day, or, since a gallon of water weighs 10 lb., 2,240,000,000 lb. of water, i.e. 1,000,000 tons per day. Of this huge quantity 130,000,000 gallons per day are drawn from the River Thames—not from the Thames at the heart of London, but at points just inside and, indeed, just outside the boundary, as, for example, at Staines. A further 50,000,000 gallons are obtained from the River Lea, while from wells sunk down to the chalky strata that underlie the Metropolis a further 44,000,000 gallons are obtained. Some idea of the pumping machinery required to deal with London's water supply can thus be gathered. But there is also the drainage problem, i.e. dealing with the rainfall and the sewage; and while to a very large extent this is assisted by gravitation, the water flowing naturally to the lower levels towards the mouth of the river where it is discharged, pumps are required to deal with heavy rainfall and also to assist the flow of the sewage water. For this purpose in various parts of London eleven pumping stations have been installed which in the aggregate are capable of dealing with no less than 60,000,000 gallons of water per hour, and even this is sometimes found to be inadequate during long periods of exceptionally heavy rainfall. The above figures give some idea of the magnitude of the problem of dealing with the needs of a large and thickly populated area like London, and of course in every town a problem of the same nature, although not perhaps of such large magnitude, has to be solved. It is beyond the scope of this work to deal with the various methods adopted for ensuring an adequate supply of water to the inhabitants of our large towns, but it can be asserted that in almost every case the pump plays a very important part, if not an essential part, in the scheme.

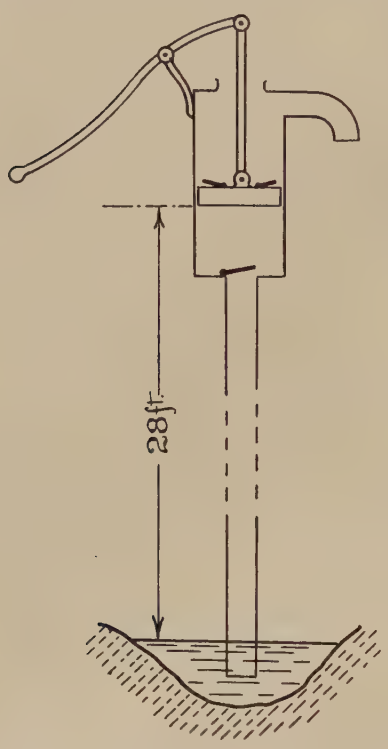


FIG. 78.—Diagrammatic Section of the Common Suction Pump.

Before, however, we proceed to describe in detail some of the different forms of pumps which are employed for dealing with these enormous volumes of water, it will be advantageous if we first of all give a brief description of the principles underlying the operation of these pumps. One of the simplest forms of reciprocating pump that can be imagined is the common suction pump such as is found at innumerable wells in country districts. A diagrammatic sketch of this pump is shown in Fig. 78, from which its action can be readily understood. The foot of the pump cylinder and also the pump plunger, or bucket as it is frequently termed, are provided with valves, as indicated, which open upwards, while from the bottom of the pump cylinder a pipe is led which dips into the water below. Imagine that the cylinder is full of water with the plunger at

the top of its stroke. If now the plunger moves downwards, the pressure on the water closes the valve at the base of the cylinder and the water passes through the plunger. But as soon as the plunger begins to move upward, the weight of water closes the valves in the plunger and the water is carried on the plunger and lifted to the level of the outlet pipe, where it is discharged. Meanwhile the upward movement of the plunger tends to create a vacuum in the lower portion of the cylinder, as a result of which the pressure of the air acting on the water in the well will cause it to rise in the suction pipe, thus keeping the cylinder full, so that on the next downward stroke water will pass through the plunger and be discharged at the outlet as before. It must be noted, however, that no matter how accurately the plunger may fit in the cylinder, even if it were possible to create a perfect vacuum in the suction pipe, it would be impossible for the pressure of the atmosphere acting on the surface of the water below to support a column of water of a greater height than 34 ft. The pressure of the atmosphere is about 14·7 lb. per sq. in. and this is, as we have already explained in that section of our work dealing with steam condensers,[1] equivalent to a column of mercury about 30 in. high as in the mercury barometer, or to a column of water 34 ft. high. Actually, of course, we cannot hope to produce a perfect vacuum below the plunger of our pump, and thus the maximum depth from which water can be raised will be somewhere in the region of 28 ft. It will be realised, therefore, that even with the most perfect form of suction pump it is impossible to lift water through a greater distance than, say, 30 ft.

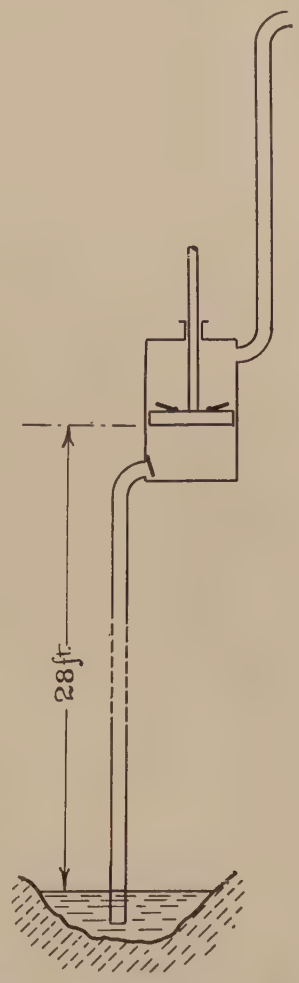


FIG. 79.—Diagrammatic Section of the Common Lift Pump.

It very frequently happens, however, that we desire to lift water from a far greater depth than 30 ft., as, for example, when it is necessary to keep a coal mine free from water, or to obtain water for domestic purposes from a deep well. Hence, unless we are prepared to take the pump with its driving engine down to within 30 ft. of the free surface of the water, some other method must be adopted. Under these circumstances a lift pump is employed. The difference between this type of pump and the ordinary suction pump just described will be at once understood from a glance at Fig. 79, which shows in diagrammatic form a simple lift pump. The action in the cylinder is precisely the same as in the suction pump, but the water, after it has passed the valves in the plunger, is lifted in the delivery pipe to the desired height at which discharge occurs. Thus, while the distance between the pump cylinder and the free surface of the water is again not more than about 28 ft., the lift which can subsequently be given to the water depends only upon the power behind the piston and its strength to resist the pressure due to the weight of column of water above it. If the height of this column is considerable, and we have the engine which drives the pump at the top of the well, it is apparent that a very long pump rod will be required, and the weight of this rod together with the weight of the piston may be very great indeed. Nevertheless, with certain modifications, this type of pump

[1] See Volume I, Chapter IX.

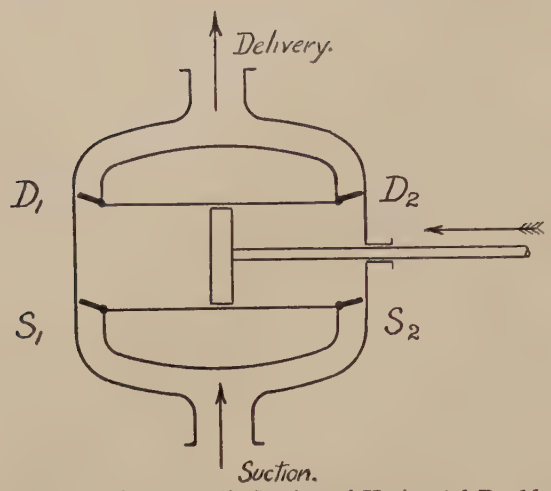


FIG. 80.—Diagrammatic Section of Horizontal Double-acting Pump.

with its long rod is very frequently used in the pumping stations of our large waterworks.

A very common method of supplying water to a town is to construct reservoirs at suitable positions on the highest points available near the town, so that the water to the houses flows by gravity. It is, of course, necessary to keep the reservoirs full, and to do this, water has to be pumped usually from the source of supply, which may be deep wells or rivers, to the reservoirs. Under these circumstances the pump itself may be immediately adjacent to its engine so that the long piston rod just referred to is dispensed with. Moreover, the pumps are usually designed with their cylinders horizontal, and the pump plunger is driven by a continuation of the piston rod of the engine. The pump already referred to in Plate III is an example of this type. It is to be noted, however, that pumps of this type are usually made double acting—that is to say, water is delivered both on the forward and on the return stroke of the plunger and not merely on the upward stroke, as in the case of the simple suction and lift pumps just described. The action of the horizontal double-acting pump will be readily understood from Fig. 80, which shows such a pump in simple diagrammatic form. It will be noticed that at each end of the cylinder there are two valves, one on the suction pipe and one on the delivery pipe, and that these valves all open upwards as indicated. Imagine now that the pump cylinder is full of water. As the plunger moves backward in the direction of the arrow, the tendency to increase the pressure on the left-hand side of the plunger closes the valve S1, but lifts the valve D1, through which the water is forced. Meanwhile there is a tendency to create a vacuum behind the plunger, i.e. on the right-hand side, and thus, provided the length of the suction pipe does not exceed about 28 ft., water passes through the valve S2, the weight of the water in the delivery pipe closing the valve D2. This action on both sides of the plunger continues until the end of the stroke, when as soon as the plunger begins to move in the opposite direction, the action on each side of the plunger is reversed, the valve S1 opens and D1 shuts, while S2 shuts and D2 opens, so that water is delivered from the space on the right of the plunger while the space on the left is being filled from the suction pipe.

Even with the double-acting pump, while we get a certain volume of water discharged at each stroke of the plunger, we do not get a continuous discharge of water, since both at the beginning and at the end of its stroke the plunger is at rest, while its velocity, and with it the velocity of the water discharged, varies throughout the entire stroke, becoming a maximum towards the middle of the stroke. In order to obtain a more or less uniform rate of discharge from the pump, it is usual to fit one or more air vessels along the delivery pipe. The function of these air vessels is to act

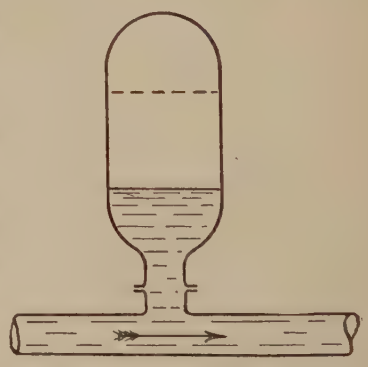


FIG. 81.—Diagrammatic Section of Air Vessel on Pump Delivery Pipe.

as small reservoirs for the water which is delivered when the plunger is moving with its maximum velocity, delivering this water to the main when the piston is at the end of its stroke. These air vessels, which form a prominent feature of all pumps, are generally somewhat of the form shown diagrammatically in Fig. 81. When the pump is discharging water at its minimum velocity the level in the air vessel will be approximately as shown by the full line ; but as the velocity increases, the pressure in the main also tends to increase, with the result that the air in the vessel is compressed and the water rises to the position shown by the dotted line, a certain volume of water being thus stored in the vessel. Then, as the velocity of the piston decreases, the pressure falls and the water in the vessel is discharged to the delivery main. In this simple way depending on the size and number of the air vessels, a more or less uniform rate of discharge is maintained. In all systems of water pipes it is advisable that the velocity of the water should be kept as uniform as possible, otherwise vibrations are liable to occur, particularly at bends and valves, which may have very serious consequences, leading ultimately to fracture of the pipe. The presence of air vessels in the pipe line tends to eliminate the possibility of this.

It will be apparent that the capacity of a pump depends not only upon the number of gallons of water which it discharges in a given time, but also upon the height to which it lifts the water. Thus the pump shown on Plate III is described as having a capacity of 4,500,000 gallons per day against a head of 360 ft. The term " head " is employed to indicate the difference of level between the point at which the water is drawn into the pump and the point at which it is discharged. Frequently the suction head, i.e. the lift on the suction side, is so small compared with the delivery head, i.e. the height to which the water is lifted on the delivery side, that the former is not taken into account, and the lift then becomes the head against which the pump works. In any case, when once we know the amount of water delivered by the pump and the height to which the water is lifted, we can at once determine the horse power of the pump. Thus, if the pump delivers N gallons of water per day, since a gallon of water weighs 10 lb., the weight of water raised per day is 10N lb., or the weight raised per minute is $\frac{10N}{24 \times 60}$ lb. If now this water is raised through a head h ft.—h being the suction head (if any) *plus* the delivery head—then the work done per minute by the pump is

$$\frac{10N}{24 \times 60} \times h \text{ ft.-lb.}$$

and since 1 h.p. is equivalent to 33,000 ft.-lb. of work per minute, we have

$$\text{Horse power of pump} = \frac{10N}{24 \times 60 \times 33{,}000} \times h.$$

There will, of course, be a certain amount of loss of power in the pump itself due to friction of the piston, work done in operating the valves, etc., so that if η is the efficiency of the pump, i.e. the ratio of the energy got out of the pump to the energy supplied by the engine, the horse power of the engine which drives the pump will be given by

$$\text{Horse power of engine} = \frac{10N}{24 \times 60 \times 33{,}000} . h . \frac{1}{\eta}$$

By substituting the figures given for the pump shown on Plate III, it will be found that the horse power of the pump itself is $\frac{10 \times 4{,}500{,}000}{24 \times 60 \times 33{,}000} \times 360 = 340$ h.p., while, assuming that the efficiency of the pumps is 95 per cent., the horse power delivered by the engines is $\frac{340}{\cdot 95} = 360$ h.p.

Another method whereby the output of a pump is sometimes expressed is to state the pressure of the water in the delivery pipe in addition, of course, to the quantity of

water delivered. Thus, when referring to boiler feed pumps, we do not speak of the head against which the feed water is delivered, but of the working pressure in the boiler. Moreover, with many types of pumps, instead of giving the capacity in gallons per hour, the output is expressed in tons per hour. In both cases the units can be readily converted from one system to the other. Suppose we are dealing with a pump delivering water against a head of h ft. and we desire to express this as a pressure in lb. per sq. in. We know that 1 cu. ft. of water weighs 1,000 oz., i.e. 62·5 lb., and hence a column of water 1 sq. ft. in cross-sectional area and 1 ft. high would exert a pressure of 62·5 lb. on the base of the column. If the height of the column is 2 ft., the pressure at the base is doubled, since 2 cu. ft. of water are now supported by the base of the column. Similarly, if the height is h ft. the pressure at the base is 62·5 h lb. per sq. ft. It is not necessary, however, for our column to have a cross-sectional area of 1 sq. ft.; no matter what the area may be, the pressure on each unit of area will be the same for a given height of column, and thus if, instead of using the figure 62·5 lb. as above, we take w as the weight of 1 cu. ft. of the liquid being pumped and h the height through which it is being delivered, the working pressure in the pump $p = wh$. Thus, in the case of the pump delivering against a head of 360 ft. the pressure is $62{\cdot}5 \times 360 = 22{,}500$ lb. per sq. ft., or since our pressures are usually expressed in lb. per sq. in., we have $\frac{22{,}500}{144} = 156$ lb. per sq. in. Hence the working pressure in the cylinder of the pump is 156 lb. per sq. in. To convert the output in gallons to tons, we have 1 gallon of water weighs 10 lb., therefore 224 gallons weigh 2,240 lb., i.e. 1 ton, so that the capacity of the pump in gallons divided by 224 gives the output in tons, or *vice versa* the capacity in tons multiplied by 224 gives the output in gallons. It may frequently be necessary to make use of these simple rules for expressing the output of a pump in any desired units.

In our brief historical survey of the development of the steam engine [1] mention was made of the fact that the earliest forms of steam engine were utilised to pump water from the mines, and it was primarily owing to the necessity of finding more and more powerful pumps for keeping the mines free from water as the workings became deeper and deeper that the early developments in the steam engine were made. Savery's engine, Newcomen's engine, and Watt's early engines were all employed to drive pumps, and while the engines of Watt and his contemporaries were gradually adapted to perform other kinds of work—driving machinery in mills, locomotives, and steamships—in its earlier forms at least the steam engine was mainly used for driving pumps. These earlier engines, as we have seen, were of the beam pattern, the steam cylinder being mounted on one side of the beam and the pump cylinder on the other, so that as the reciprocating motion of the piston was given to one end of the beam, its motion was transferred to the pump plunger connected to the other side of the beam. The reader is referred to the drawings of these earlier forms of engine, reproduced in the chapter referred to above, when the principle employed will at once be apparent. So simple and so satisfactory is this type of drive for pumps that it is not surprising to find it still extensively employed at the present day. In many waterworks pumping stations all over the country these beam engines are to be found performing their work efficiently and well, some of them being of very great age, and others comparatively new. The desire to use compound or even triple-expansion steam engines makes it rather a difficult matter to employ the beam principle, but even at the present time makers are occasionally called upon to build engines of this type and in these the compound principle is generally employed.

As a modern example of a beam engine, we may consider Fig. 82, which shows one of a pair of compound condensing beam engines designed and built by Messrs. Glenfield & Kennedy, of Kilmarnock, to the order of the Maidstone Waterworks Co. The engine

[1] See Volume I, Chapter III.

is shown during erection in the makers' shops. The cylinders, which are steam-jacketed, are shown on the left, the diameter of the high-pressure cylinder being 18½ in., and that of the low-pressure cylinder 30 in., the stroke of the pistons being 3 ft. and 4 ft. 6 in. respectively. It is hardly necessary to point out that with the high-pressure cylinder arranged nearer the centre of the beam the stroke of its piston must be less than that of the piston in the low-pressure cylinder, which is arranged at the end of the beam. The

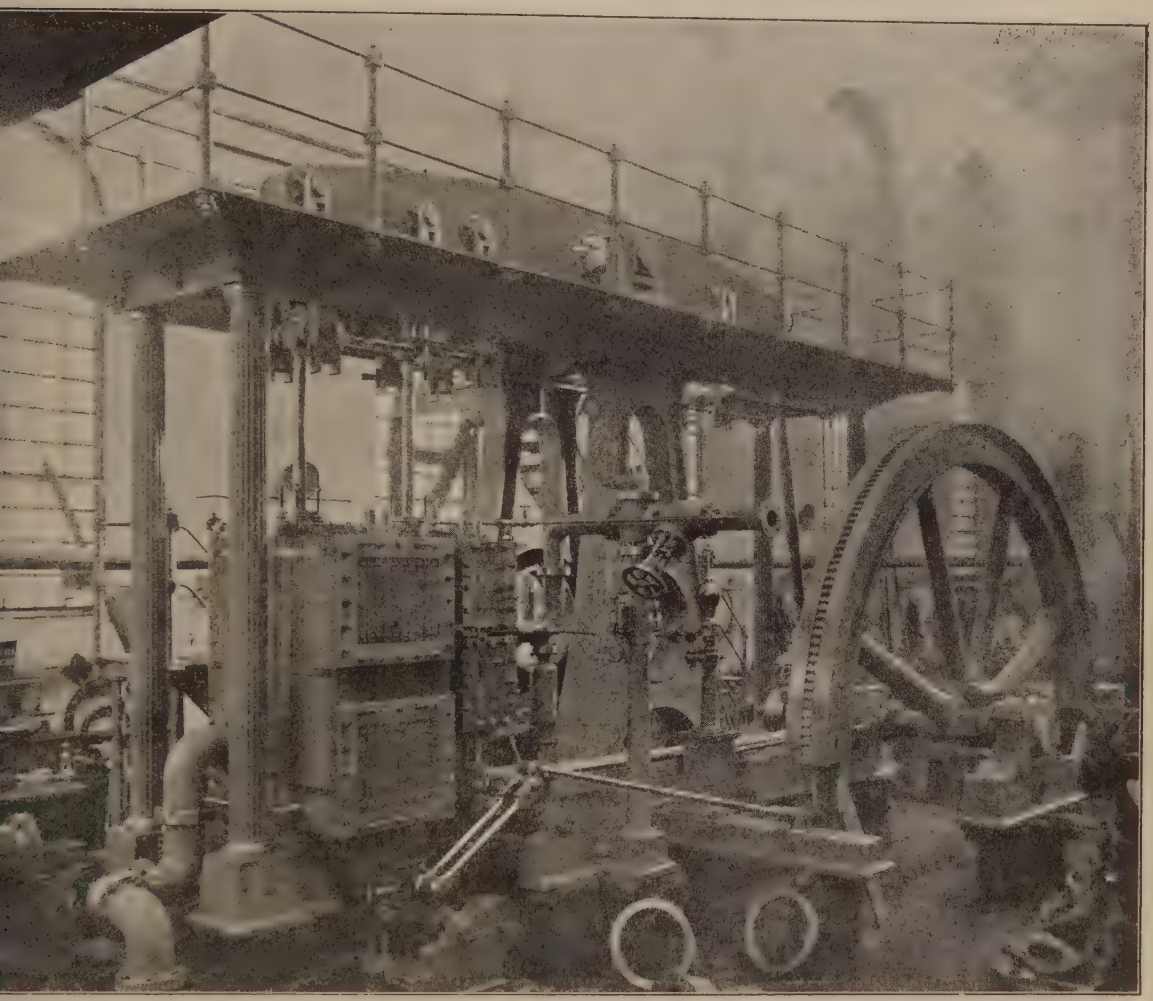


FIG. 82.—Compound Beam Engine by Messrs. Glenfield & Kennedy, Ltd., Kilmarnock, driving Two Double-acting Pumps of total capacity 1,250,000 gallons per day, against a total head of 430 ft. The steam cylinders are on the left.

engine drives two double-acting pumps each 11 in. diameter with plunger strokes of 26 in. and 33 in. respectively, while in order to give smooth and uniform running, a connecting rod secured to the other end of a beam drives a crank shaft carrying a heavy flywheel as shown. The engine is designed for a speed of 24 revs. per min., using boiler steam at a pressure of 80 lb. per sq. in. Under these conditions the pumps are capable of raising 1,250,000 gallons of water per day of 24 hours against a total head of 430 ft. The beam is built up of double steel plates with cast-iron distance pieces securely riveted between them. Both piston rods are guided vertically by an improved design of Watt's

parallel motion, a description of which has already been given in conjunction with our description of Watt's engines in the chapter referred to above.

The general lay-out of a modern beam engine together with its pumps and the

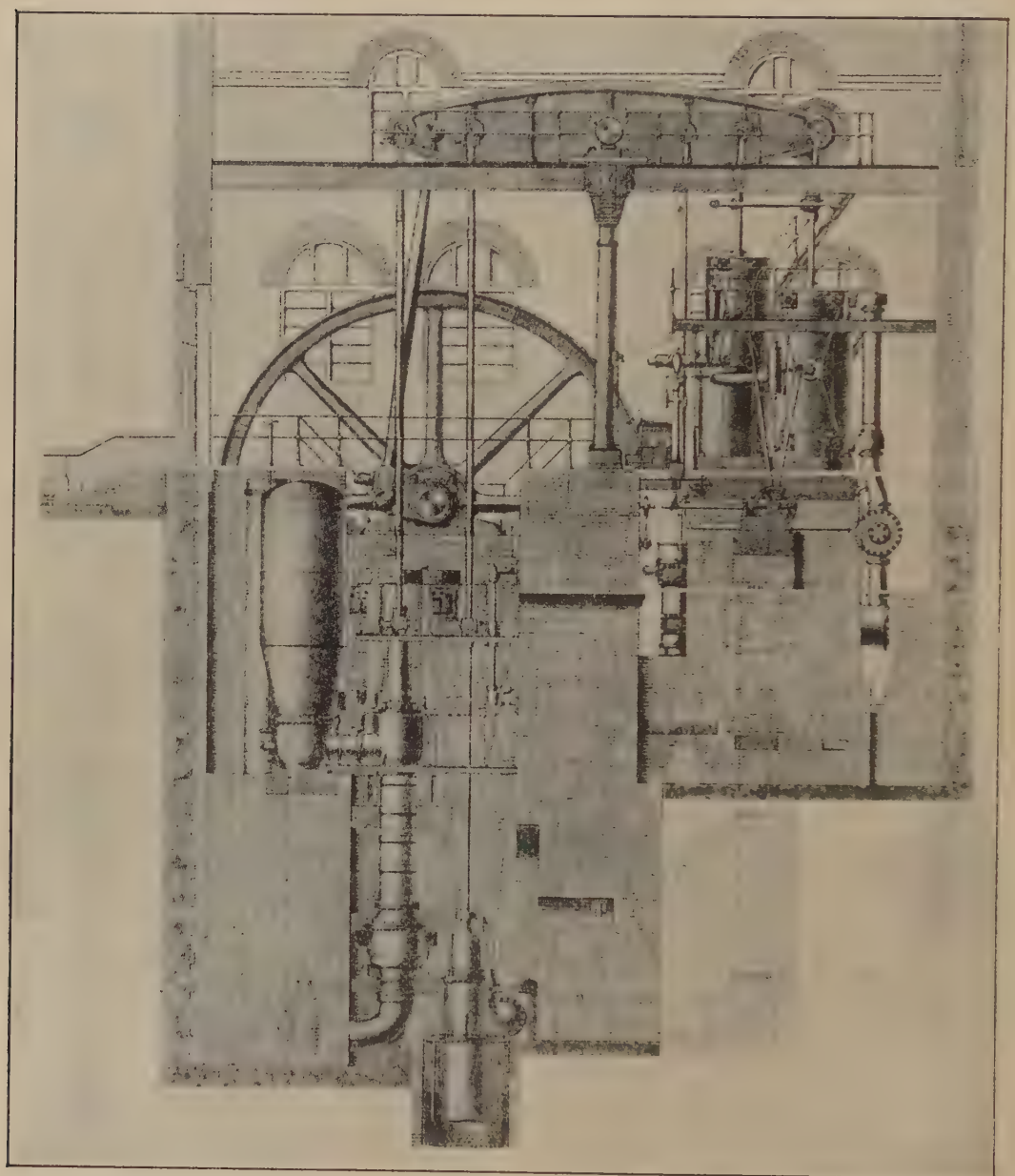


FIG. 83.—General Lay-out of Compound Beam Engine driving a Combined Deep Well and High Lift Pump. Capacity 3,300,000 gallons per day; total lift 380 ft. Designed and constructed by Messrs. Worthington-Simpson, Ltd., Newark-on-Trent.

arrangement of the foundations is clearly shown in Fig. 83, which shows a unit designed and constructed by the well-known firm of pump makers, Messrs. Worthington-Simpson, Ltd., whose works are at Newark-on-Trent. This unit, which was built and installed to the order of the Brompton, Chatham, Gillingham, and Rochester waterworks, is designed

to raise $3\frac{1}{3}$ million gallons of water per day from a deep well and deliver it into a high service reservoir, the combined lift being 380 ft. The engine, which is on the right, is of the compound surface condensing type, both the cylinders being steam-jacketed, while in order to further improve the efficiency of the engine, the receiver between the high- and the low-pressure cylinders is fitted with tubes through which steam at boiler pressure is passed, this acting as a reheater for the steam entering the low-pressure cylinder. The distribution of the steam is carried out by means of Corliss gear as shown, the valves in both cylinders being driven independently from a central wrist plate actuated by eccentrics which are mounted on a countershaft as shown. The cut-off valves are fitted with "trips," those on the high-pressure cylinder being controlled by a sensitive high-speed governor, those on the low-pressure cylinder being adjustable by hand while the engine is running. The speed of the engine can, however, be altered if necessary while the engine is running by means of a wheel adjustment on the governor, and there is also a safety device incorporated with the governor whereby, should the speed of rotation vary within a close percentage above or below the predetermined normal, the high-pressure valves are thrown out of action and left in a position covering the steam inlet ports, while at the same time a valve on the condenser is opened and the vacuum is destroyed. In this way the engine is brought to rest and the conditions which led to the fluctuation of speed can be investigated.

From the left-hand side of the beam two pumps are driven, the one a deep well pump which draws the water from the bottom of the well and delivers it to a tank at the ground level, and the other a high lift pump which takes the water from this tank and delivers it to the surface reservoir. The deep well pump will be observed nearer the fulcrum of the beam, the top of the well being shown in the drawing, the pump cylinder with its delivery pipe being included although the pump itself is not drawn. The deep well pump is of the usual plunger type with long pump rod as already described. The high lift pump is also of the plunger type. In order to give steady running, a heavy fly-wheel is fitted, this being driven by a connecting rod operated from the left-hand side of the beam as shown. The reader will be able to determine the horse power of the pump from the simple formula already given. It will be found that this works out to 266 h.p. The makers specify that the power developed by the engine is 300 h.p. when steam at 100 lb. sq. in. boiler pressure is supplied. Thus the efficiency of the pump is $\frac{266}{300} = 88{\cdot}6$ per cent. It is further specified that the plant will maintain a duty of 125,000,000 ft.-lb. of work in normal running per cwt. of Welsh coal. This is equivalent to raising $\frac{125{,}000{,}000}{10 \times 380} = 33{,}000$ gallons of water through a total head of 380 ft. for each cwt. of coal burned if good Welsh coal is employed.

It has already been noted that the beam engine is gradually giving place to other forms owing to the fact that it is rather a difficult matter to arrange for three expansions in cylinders having different piston strokes, and also because of the long length of beam that would be required if these cylinders were arranged side by side beneath one half of the beam. Judging from a number of large pumping units installed in recent years, it would appear as though the place of the beam engine is being gradually usurped by the vertical inverted triple-expansion engine, a type of engine which is very similar in design to the reciprocating marine engine which is still so extensively employed in the mercantile marine for the propulsion of cargo steamers. An example of this type of engine has already been given in the introductory chapters of this work, the engine being illustrated in Plate III of Volume I. It will be of interest, however, to give a rather more detailed description of this engine because it is typical of the type which is being extensively employed in waterworks and pumping stations at the present

time. This engine is one of three which were designed and constructed by Messrs. Glenfield & Kennedy for the Glasgow main drainage scheme and installed by them at the Kinning Park pumping station for dealing with the sewage water after it has been finely screened. The diameter of the high-pressure cylinder is 15½ in., of the intermediate-pressure cylinder 25 in., and of the low-pressure cylinder 42 in., the pistons in all three cylinders having a common stroke of 4 ft. 6 in. The normal running speed of the engine is about 23 revs. per min. Corliss valve gear is fitted to each cylinder to give the correct distribution of the steam, the point of cut-off on all cylinders being regulated by the governor, while an automatic emergency knock-out throttle valve is also fitted on the main steam pipe which operates when the speed of the engine rises above a predetermined limit. The cylinders are all steam-jacketed, and tubular reheaters are placed in the two receivers between the cylinders. Two heavy flywheels are mounted on the crank shaft to give steady running, the one nearer the observer having teeth cut on its rim for operating with an electric barring gear while the other flywheel has square holes cast in the rim for barring by hand. The man standing in the background of the picture (Plate III, Volume I) serves to give some idea of the overall dimensions of these engines.

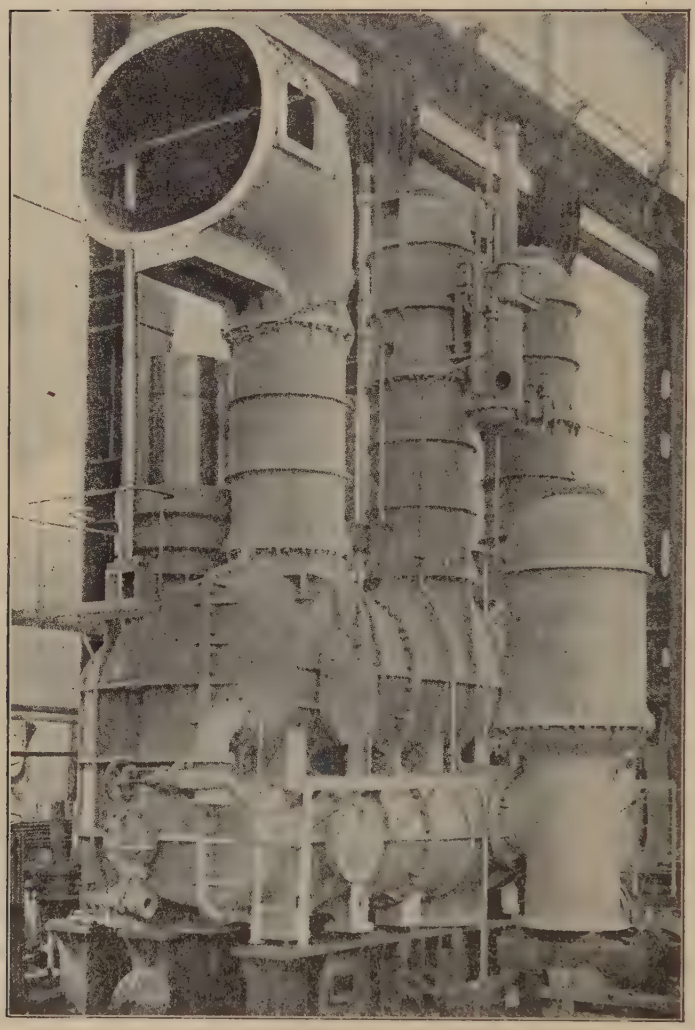


FIG. 84.—Triple Plunger Single-acting Pump by Messrs. Glenfield & Kennedy, installed at the Kinning Park Pumping Station, Glasgow.

The pumps are of the triple plunger single-acting type and are erected immediately below the engine. Each pump plunger is directly beneath one of the cylinders of the engine and is driven from the crosshead by means of four rigid rods. A view of one of the pumps as erected in the makers' shops is shown in Fig. 84, the plungers being at the back, although two of them can be clearly seen, the one on the left being completely visible. These plungers are each 3 ft. 6 in. diameter, while the stroke is of course the same as that of the engine, viz. 4 ft. 6 in. The pumps are designed to deal with 12,320 gallons of screened sewage per minute, raising it through a total head of 46 ft., which

is equivalent to 17,740,800 gallons per day of 24 hours, the weight of sewage raised being from 58 to 60 tons per minute, which gives a horse power at the pump of about 187. Moreover, the makers specify that the efficiency of the engine and pump is equal to 92 per cent., so that the power developed by the engine is about 205 i.h.p. The pump valves are of the hinged flap pattern faced with leather, the flaps themselves being of cast steel hinged on to cast-steel frames. The pump chambers are of cast iron resting on cast-iron girders as shown. There is a vacuum vessel on the suction pipe close up to the pump chambers, and also an air vessel on the pumps under the high-pressure and intermediate-pressure cylinders. The discharge branch is on the pump under the low-pressure cylinder, and the filtered sewage passes wholly and directly through the tubes of the surface condenser. Thence it passes into the rising main, which is also provided with ample air vessel capacity to reduce the possibility of shock in the main. Sluice valves are provided so that each set of pumps can be isolated if desired. Attention may be drawn to the small pump secured at the top of the air vessel of the centre main pump. This is the air pump of the engine and it is operated by levers from the intermediate crosshead. The overall dimensions of these units will be gathered from the fact that the distance from the pump base to the engine room floor is 27 ft. 3 in., and from the engine room floor to the top of the relief valves on the cylinder covers 27 ft., making a total height of 54 ft. 3 in. from the pump base to the top of the engine.

As a further example of the application of the vertical triple-expansion engine for pumping purposes, we may consider the units installed by Messrs. Ashton, Frost & Co., Ltd., of Bank Top Foundry, Blackburn, at the Elkesley Pumping Station of the City of Lincoln Waterworks Department. These units, of which two have been installed, are each designed to deliver 3,600,000 gallons of water per day of 24 hours against a total head of 724 ft., and consist of a vertical triple-expansion engine driving three high lift and two deep well pumps, the former being driven direct from the engine crossheads, while the latter are driven from cranks mounted one at each end of the engine crank shaft. The general lay-out of the plant will be gathered from the frontispiece to this volume, in which is shown an elevation and a plan of the complete station, the second engine being removed in the plan to show the arrangement of the pumps. As will be seen from the drawings, steam for operating the engines is supplied by three Lancashire boilers, these being each 28 ft. long and 7 ft. diameter, designed for a working pressure of 180 lb. per sq. in. The boilers are fitted with superheaters arranged at the back of the boilers as shown in the elevation,[1] and the hot flue gases after leaving the boilers pass to a Green's economiser before finally passing to the chimney. From the superheaters the steam is delivered to the engine at a temperature of about 540° F.

The engines, to which, following locomotive and steamship practice, the names JANETTA and LIVENS have been given, have cylinders 26 in., 45 in., and 68 in. diameter with a common piston stroke of 4 ft., the cranks being arranged at 120 degrees to each other, in the sequence high pressure, low pressure, and intermediate pressure, this, it is thought, yielding a more uniform turning moment on the shaft. The normal speed of the engines is 24 revs. per min. The general appearance of these engines will be gathered from Fig. 85. The three main cranks and also the rods leading down from the crossheads to the three high lift pumps will be noted, and attention may also be drawn to the overhung cranks at the ends of the crank shaft, from which the deep well pumps are driven. As will be seen, the distribution of the steam to the various cylinders is carried out by means of Corliss valve gear, the cut-off in the high-pressure cylinder being controlled by the governor, while on the other cylinders it is capable of adjustment by hand while the engine is running. All the cylinders are provided with steam jackets both on the barrel and on the covers, and the receivers between the cylinders are fitted

1 See Volume IV, Chapter I, for a detailed study of the Lancashire boiler with its superheater, etc.

with reheating tubes through which boiler steam is passed. The condenser, which is of the surface type, contains 290 solid drawn tubes 8 ft. long between the tube plates and $1\frac{1}{8}$ in. internal diameter, giving a total of 780 sq. ft. of cooling surface. The body of the condenser is placed in the delivery pipe of the deep well pumps as shown in the drawings, the cold water as drawn from the wells serving as circulating water in the condensers.

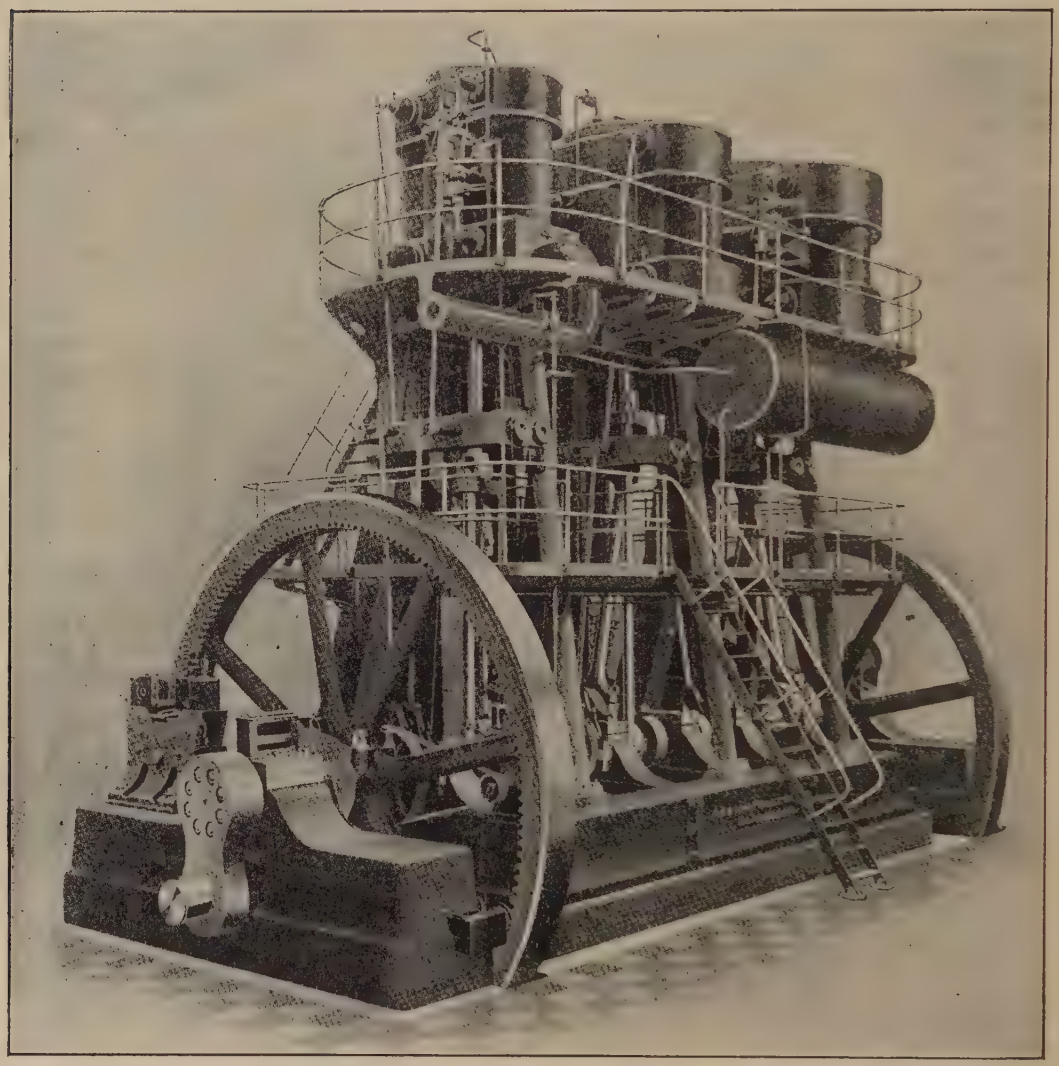


FIG. 85.—Vertical Triple-expansion Pumping Engine designed by Messrs. Ashton, Frost & Co., Ltd., and installed at the Elkesley Pumping Station of the City of Lincoln Waterworks Department.

The air pump is of the Edwards type driven direct from a bracket on the crosshead of the high lift pump adjacent to the low-pressure cylinder.

The deep well pumps are fixed at a level of 314 ft. below the level of the engine room floor, and, as already noted, they are driven by overhung cranks fitted to the ends of the crank shaft. The radius of these cranks is 2 ft. 6 in., so that the stroke of the plunger is 5 ft., the diameter of the pump bucket or plunger being 18 in. The cranks are each a bored fit on the flange, which is forged on to the crank shaft, and the cranks are secured to these flanges by means of eight turned and fitted bolts. By this arrangement

the crank can be removed from the shaft when it is desired to lift the pump buckets and the foot valves for examination and overhaul. The deep well pumps deliver the water through the condensers into a settling tank, from which it flows over a weir into a suction tank common to the three high lift pumps. These pumps, which are placed vertically below the engine cylinders, are driven from brackets forged on the piston rod crossheads by two strong side rods. The plungers are 16¼ in. diameter, and the stroke is, of course, the same as that of the engine, viz. 4 ft. The general arrangement of these pumps will be understood from Fig. 86, which is a view showing the air vessels, with the pump cylinders in the background. The plunger of one of the pumps can, however, be clearly seen, as well as its stuffing box. It will be noticed that this is provided with a toothed ring, and that each of the gland nuts is provided with a small toothed wheel so that when one nut is turned, all the nuts must turn an equal amount, thus pressing the gland uniformly all the way round. These pumps deliver the water into a cast-iron main 21 in. bore, through which it is passed to the tank of the Westgate Tower—a distance of twenty-two miles.

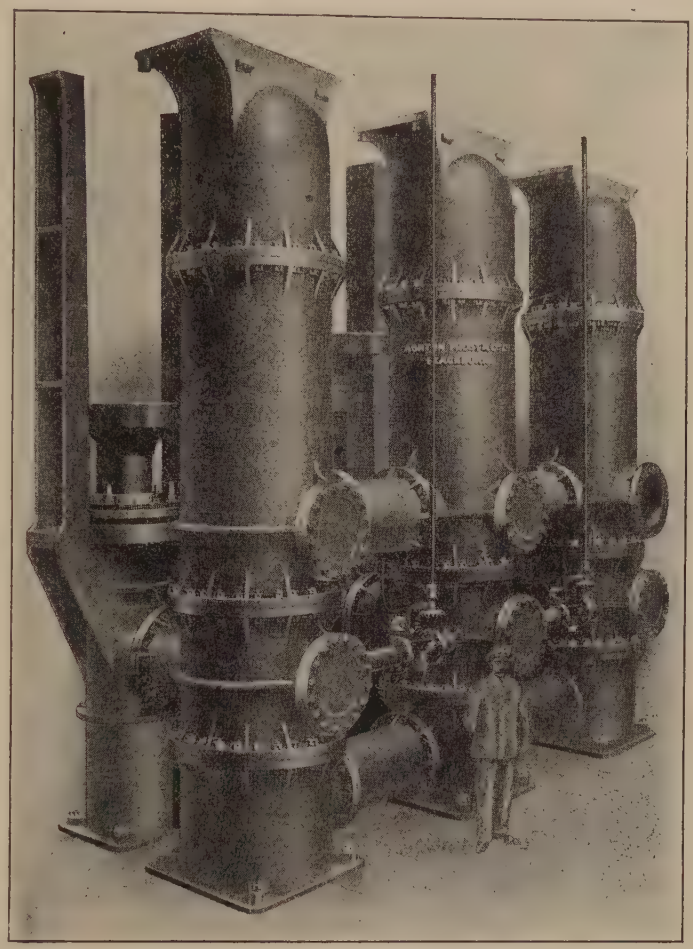


FIG. 86.—Triple Plunger High Lift Pumps delivering 3,000,000 gallons of water per day a distance of twenty-two miles.

In order to give some idea of the efficiency of these engines and of the pumps, a summary of the figures obtained during the acceptance trials may be given. After erection the engines were tested on behalf of the City of Lincoln by the British Engine, Boiler & Electrical Insurance Co., Ltd., of Manchester. The guarantees given by the contractors (Messrs. Ashton, Frost & Co., Ltd.) were based on the number of foot-pounds of work done per 1,120 lb. of steam and per 112 lb. of in coal, raising the water measured by meter at Elkesley from the bore holes and delivering it into the rising main against a total head of 524 ft., while the engines were pumping at the rate of 3,000,000 gallons per 24 hours, and the boilers were being fired with coal having a calorific value of 14,000 B.Th.U. per lb. The results of the official tests which are given on the following page have been calculated in terms of these guarantees.

During these trials, the pressure of the steam on the engine side of the stop valves averaged 172 lb. per sq. in., and its temperature was 536° F., while the vacuum in the

SUMMARY OF RESULTS OF OFFICIAL TESTS OF PUMPING ENGINES AT THE CITY OF LINCOLN WATERWORKS

	"Janetta"	"Livens"
Name of engine		
Duration of trial	12 hours	12 hours
Indicated horse power .i.h.p.	403·1	385·2
Pump horse power calculated from water delivered into main as registered by meter at Elkesley and total head on pumps p.h.p.	355·2	351·9
Mechanical efficiency of engines and pumps reckoned on water metered per cent.	88·12	91·35
Total weight of steam used reckoned on condensed steam leaving engines lb.	47,412	45,504
Total weight of dried fuel fired lb.	5,170	4,936
Average temperature of hot well ° F.	113·9	102·5
Actual temperature of feed to economiser ° F.	54	54
Steam per i.h.p. hour lb.	9·80	9·843
Steam per p.h.p. hour lb.	11·32	10·97
Duty of engines per 1,120 lb. of steam. ft.-lb.	195,300,000	201,790,000
Duty of engines per 112 lb. of dried coal (actual) ft.-lb.	184,600,000	187,745,000
Equivalent duty of engines with coal having a calorific value of 14,000 B.Th.U. per lb. and feed to economiser at hot well temperature ft.-lb.	186,000,000	185,600,000

condensers was maintained at an average figure of 28¼ in. Throughout the trials indicator diagrams were taken on all cylinders, and these were used not only to determine the indicated horse power of the engines, but also to analyse the condition of the steam during its passage from the stop valve to the condenser. For this purpose, the diagrams were reduced to a common base of cylinder volumes instead of piston stroke as in the actual diagrams, the vertical ordinate representing pressure to a uniform scale. The result is a diagram equivalent to that which would be obtained if the complete expansion occurred in the low-pressure cylinder. Then by comparing the actual expansion line with the theoretical line of expansion, the dryness fraction at every point

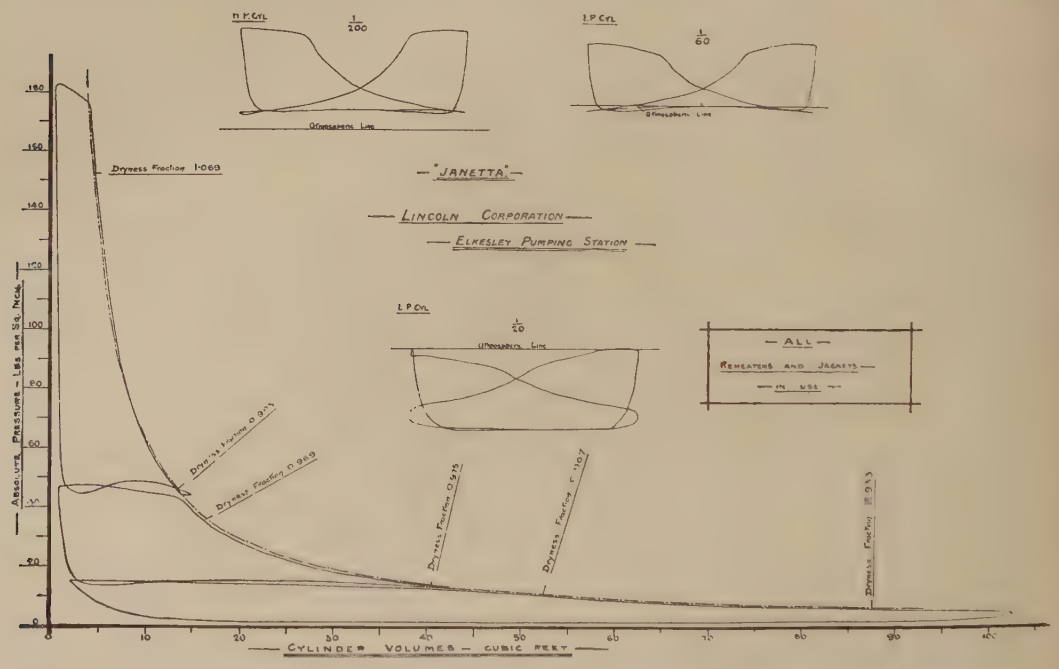


FIG. 87.—Indicator Diagrams taken during Trials of Triple-expansion Pumping Engine, reduced to Common Base of Volume of Low-pressure Cylinder.

can be readily determined. A set of these indicator diagrams is reproduced in Fig. 87, together with the diagram obtained by reducing these to a common base. It will be observed that the diagrams were taken when the jackets and the reheaters were in use, and the result of this is apparent from the fact that throughout its expansion the steam

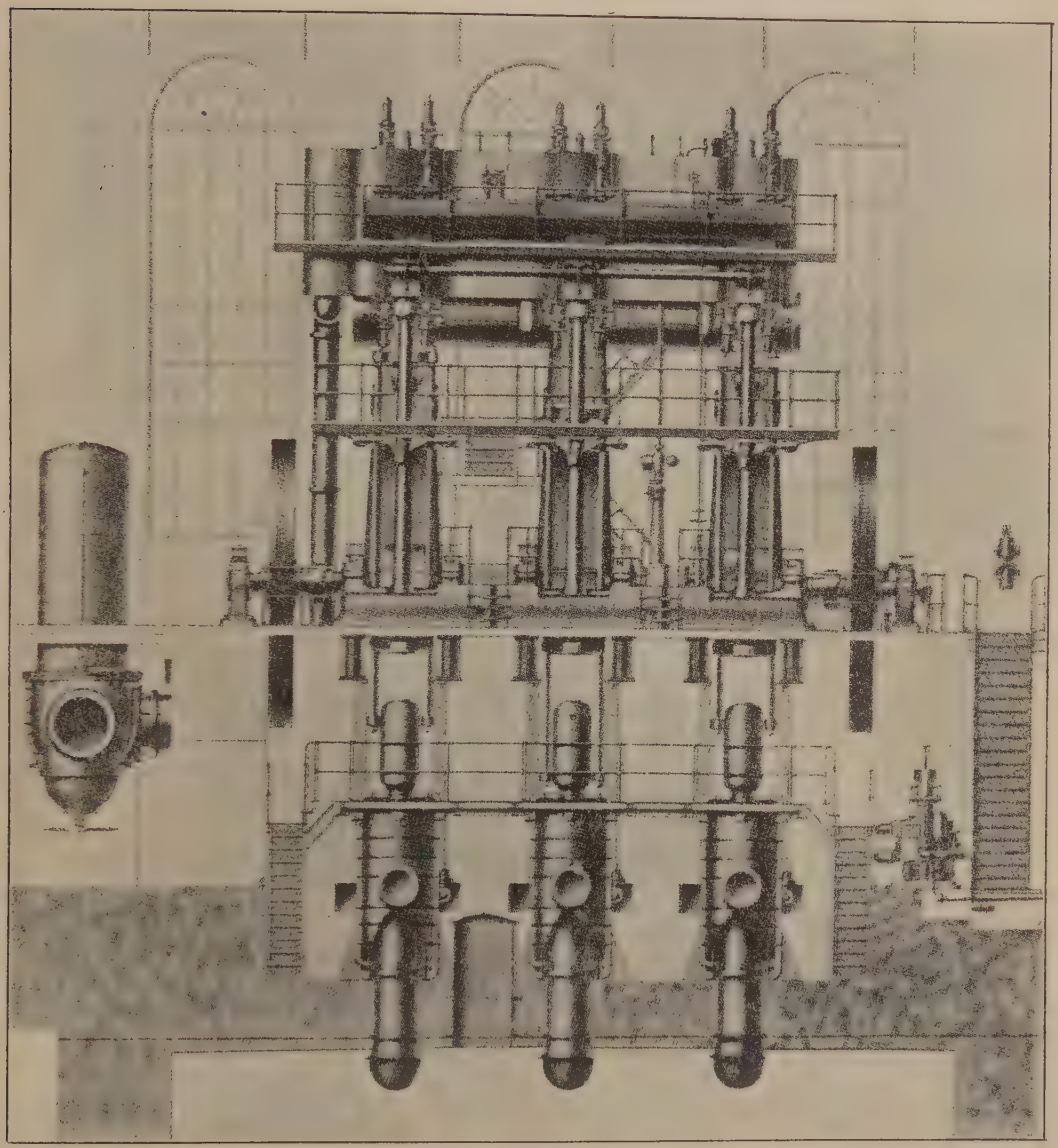


FIG. 88.—Vertical Triple-expansion Engine driving Double-acting Pumps delivering 7,500,000 gallons of water per day. Designed and constructed by Messrs Worthington-Simpson, Ltd., and installed at the Hammersmith Pumping Station of the Metropolitan Water Board.

is kept practically 100 per cent. dry. The effect of this on the efficiency of the engine is apparent. It is of interest to note that in their report on the test figures, the British Insurance Co. make the following observations: "These figures are remarkably good, and constitute, so far as the company have been able to ascertain from published results of similar tests, a record for steam-driven pumping engines at waterworks." As

a consequence of the very satisfactory results obtained, the Contractors were awarded a bonus of £1,000, the maximum possible under the contract.

While the units described above may be regarded as representing average practice as regards capacity and lift, they by no means represent the largest sizes in which pumps are constructed at the present time. As an example of a pump of very large capacity attention may be drawn to the unit shown in Fig. 88. This is a reproduction from a drawing of one of two units, each having a capacity of seven and a half million gallons per day, installed at the Hammersmith Pumping Station of the Metropolitan Water Board, the engines as well as the pumps having been designed and constructed by Messrs. Worthington-Simpson, Ltd. The engine is of the vertical triple-expansion open-fronted type with three pumps of the direct lift double-acting type arranged vertically beneath the three cylinders, the pump plungers being driven by rigid rods connected to the engine cross-heads in very much the same way as in the pump previously described. The main crank shaft carries two flywheels, one at each end of the shaft, as shown. The capacity of the pumps is 7,500,000 gallons per 24 hours, the power required for lifting the water being about 500 h.p., from which by a simple calculation it will be found that the average head against which the pumps work is about 320 ft. A notable feature of these engines is the arrangement made for the distribution of the steam, this being effected by means of double-beat drop valves of the type described in the preceding chapter. The admission valves are operated through trip gear, and the exhaust valves through positive gear, as is usual with drop valve engines. The cut-off of the steam to the high-pressure cylinder is controlled by the governor, which is also provided with an arrangement whereby the speed of rotation of the engine can be adjusted as required when the engine is running. In addition a safety governor is fitted which automatically stops the engine should the load become excessive or should it be reduced below a certain limit.

Another type of steam engine which is very widely employed for driving large pumps is the horizontal steam engine such as we have described in the preceding chapter. It will be at once obvious, from a glance at the horizontal pumping set shown in Plate III, that for a given power this form of engine occupies more ground space than does the vertical type, which means, of course, that a larger engine room will be required, involving greater rent and establishment charges, particularly if the pumping station is situated in the heart of a town where land is valuable. But against this disadvantage must be set the advantage of having all the parts of the engine easily accessible, and further that the unit possesses far greater rigidity since it is supported over a far wider area. In addition, the horizontal engine is rather cheaper to construct and install than is the vertical engine of the same power. Where pumps of moderate power are employed, the engine is usually operated on the compound principle—that is, there are two cylinders, the high pressure and the low pressure, which will be arranged either in tandem or with parallel axes according to the nature of the pumps to be driven. For large powers, the triple-expansion arrangement is adopted again with the cylinders all in line or arranged on parallel axes, while for the largest units a double triple-expansion engine is employed comprising two complete engines, each with its high-, intermediate-, and low-pressure cylinders arranged in a straight line, the two units being erected side by side and driving a common crank shaft.

Before describing some of these units in greater detail, a word must be said regarding the pumps themselves. Where high lift pumps are being driven, these can be of the horizontal type with their plungers direct coupled to an extension of the piston rod of the engine, but for deep well pumps, where the plunger may be hundreds of feet below the level of the engine room floor, this method obviously cannot be employed. In order to obtain the vertical motion for the plunger of the deep well pump from the horizontal motion of the engine, it is usual to employ some form of bell crank lever arrangement,

as shown diagrammatically in Fig. 89. An extension of the piston rod of the engine is connected to the point A of the bell crank lever AOB, while the pump rod is driven from the point B, the point O constituting the bearing of the bell crank lever. As the points A and B both describe arcs of circles about O as centre, it is necessary to employ short connecting rods between the piston rod and the pump rod respectively; the ends C and D of these rods constitute crossheads working between parallel guides, so that the piston rod and the pump rod both move in perfectly straight lines, the one horizontally and the other vertically, all pressure due to the obliquity of the connecting rods AC and BD being taken up by the guides and not by the stuffing box of the piston and pump rod. Of course, it would be possible to drive the vertical pump rod directly from a crank carried on an extension of the engine crank shaft, as is done with the vertical engines installed at Lincoln, as already described (see Fig. 85), but this arrangement is not usually adopted as the opening to the well interferes with the foundations of the main engine. For this reason it frequently happens that the bell crank lever arrangement is adopted with vertical pumping engines, the end C of the rod AC being connected to a crank on the main engine crank shaft, the rod being of far greater length relatively to the length of the arms of the bell crank lever than is shown in the diagram. In any case, it will be apparent that provided OB is of the same length as OA, the stroke of the pump plunger will be the same as that of the engine piston, although by altering the relative lengths of OA and OB, any desired stroke can be obtained.

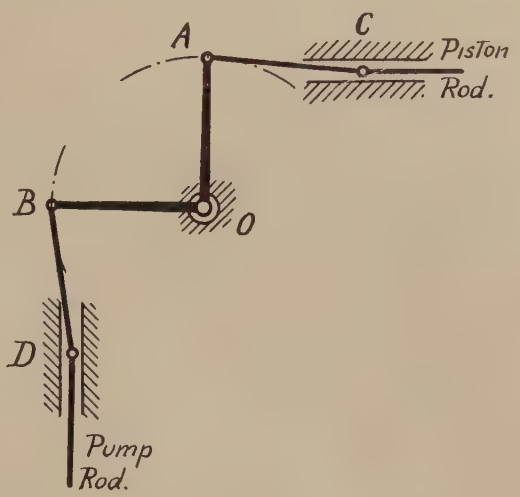


FIG. 89.—Diagrammatic Arrangement of Bell Crank Lever for driving a Deep Well Pump by a Horizontal Engine.

As an example of a pumping unit driven by a horizontal steam engine, we may consider Fig. 90, which shows a horizontal cross compound engine driving high lift and deep well pumps. This unit was designed and constructed by Messrs. Ashton, Frost & Co., Ltd., and installed by them at the Altham pumping station of the Accrington Gas and Water Board. It will be noticed that the deep well pumps, of which there are two, are driven from an extension of the high-pressure piston rod, there being two bell crank levers connected together at the top as shown. The high lift pump is driven directly from an extension of the low-pressure piston rod. The general layout of the plant will be understood from a study of Fig. 91, which shows the elevation and plan of the station. In the elevation the high-pressure cylinder is shown with the extension of the piston rod driving the two bell crank levers which operate the deep well pumps. It will be noticed that the connecting rods and also the pump rods themselves are built up of steel plates filled in with wood. At the bottom corner of the elevation on the left a sectional drawing of the pump and its plunger is given, in which the suction strainer, the foot valve, and also the plunger valve will be noticed, the valves being of the "Perniss" pyramid type. Also included in the elevation is the low-pressure engine, together with its high lift pump, this drawing being placed in the lower right-hand corner for convenience. Actually, of course, this elevation is taken immediately in front of the elevation of the high-pressure engine. It will be noticed that the high lift pump is of the double-acting type and that air vessels of large capacity are provided at both the

delivery valves. The connection between the deep well pumps and the lift pump is clearly shown in the plan. This view also serves to show the area of the well relatively to the main engines, and also the relative space occupied by the various parts of the complete unit. It will be noticed that steam for driving the engine is supplied by two Lancashire boilers fitted with superheaters.

An interesting variation of the above equipment is to be found in the Mill Mecee

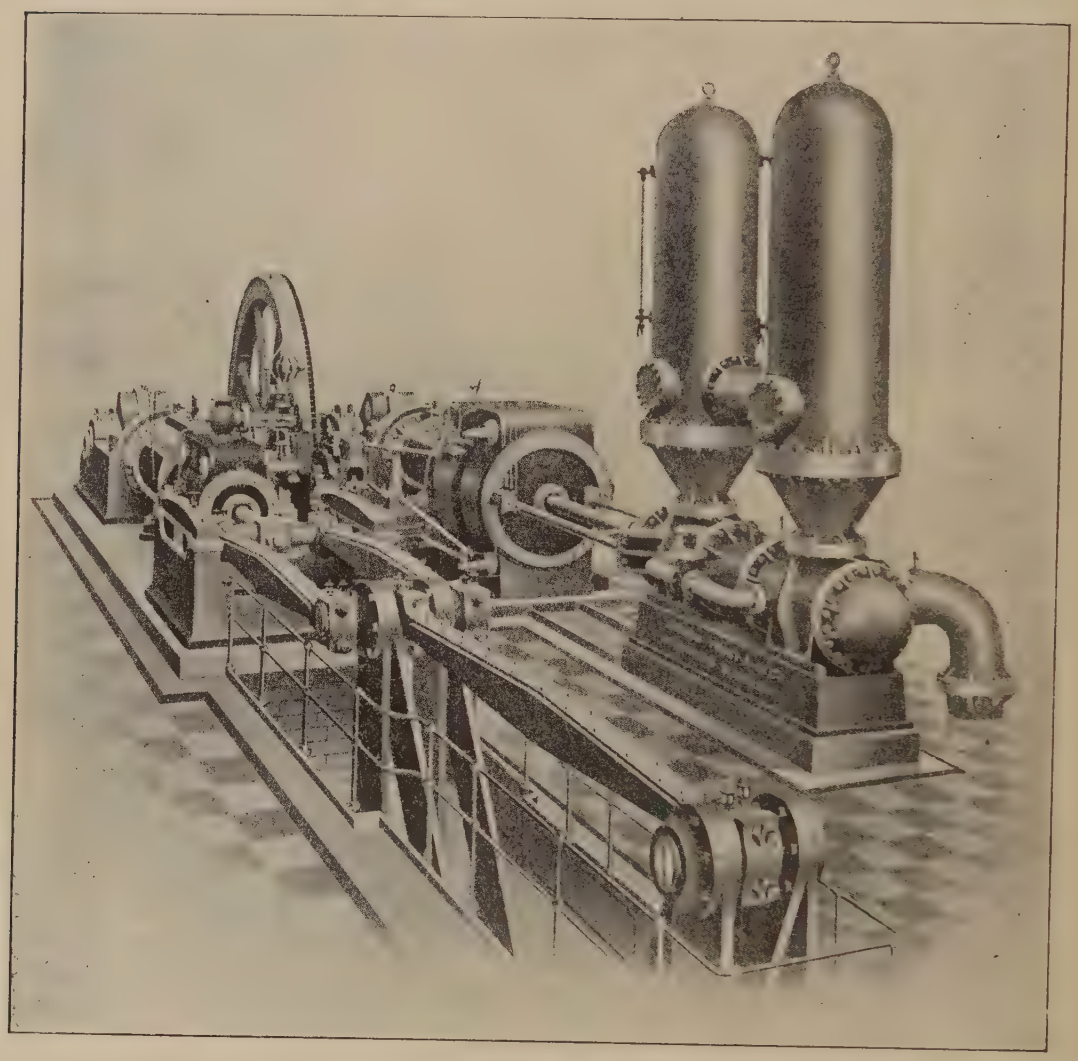
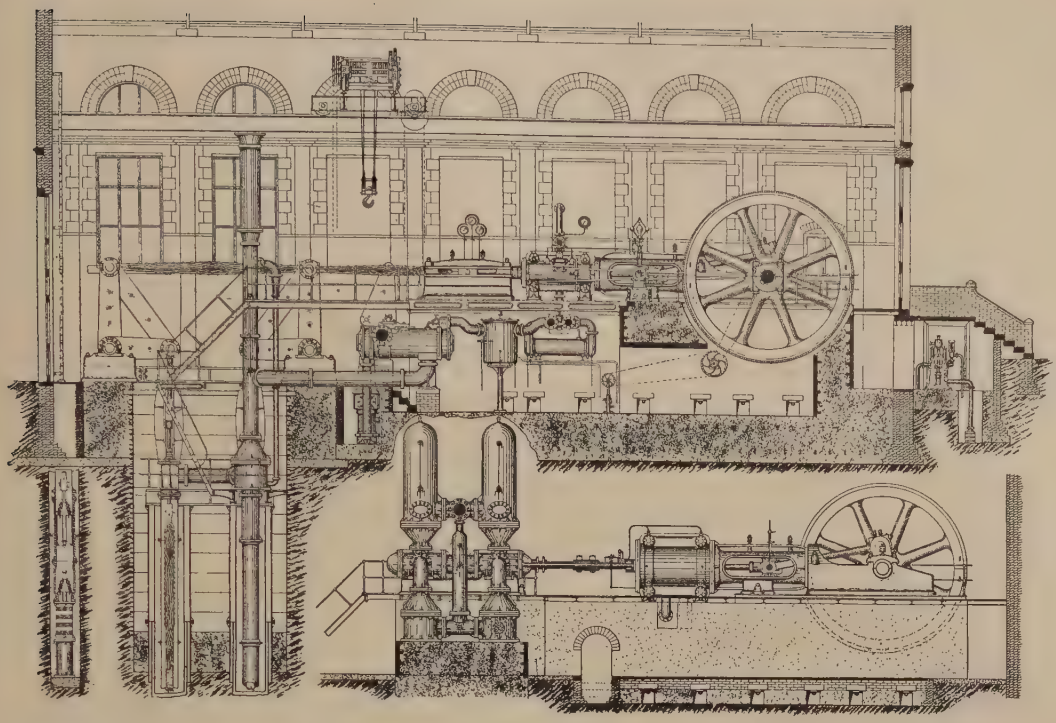
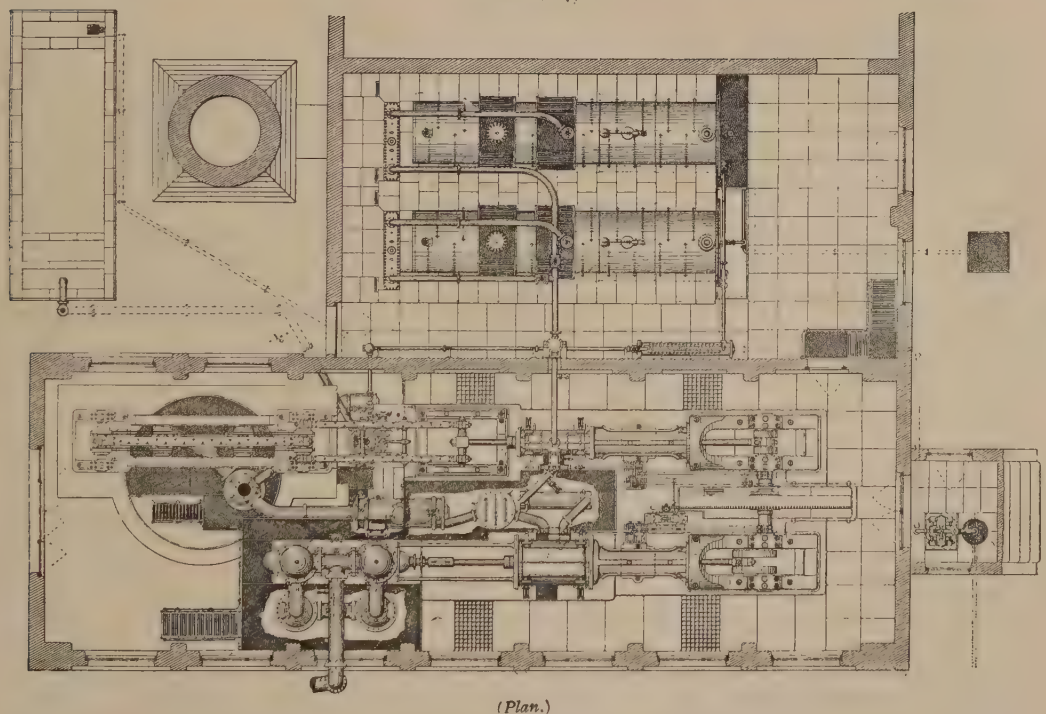


Fig. 90.—Horizontal Cross Compound Engine driving Deep Well and High Lift Pumps at Altham Pumping Station of the Accrington Gas and Water Board.

pumping station of the Staffordshire Potteries Waterworks Co., where Messrs. Ashton, Frost & Co., Ltd., have installed a tandem compound engine driving both a deep well and a high lift pump, the whole being arranged in one plane. The engine is arranged with its high-pressure cylinder nearer the main crank shaft on which is mounted a heavy flywheel to give uniform running. Immediately behind the high-pressure cylinder is the low-pressure cylinder, and from an extension of the piston rod the force pump, which is placed behind the low-pressure cylinder, is driven. Beyond this are the deep well pumps, two in number, which are driven by a continuation of the pump rod through slide



(*Elevation.*)



(*Plan.*)

FIG. 91.—General Lay-out of the Altham Pumping Station of the Accrington Gas and Water Board.

rods and bell crank levers in the ordinary way. The pumps have been designed to deliver 2,000,000 gallons of water per day of 24 hours against a total head of 604 ft. when the speed of the main engine is 18 revs. per min. The high-pressure cylinder is 26 in. diameter, and the low-pressure cylinder 49 in. diameter, the common piston stroke being 5 ft. The distribution of steam is carried out by means of Corliss valve gear, steam being supplied at a pressure of 160 lb. per sq. in. from two Lancashire boilers each 8 ft. diameter and 30 ft. long, fitted with superheaters as in the Accrington installation. The deep well pumps, which are each 15¾ in. diameter with a plunger stroke of 5 ft., are placed 139 ft. below the surface. These discharge into the force pump suction tanks, through a rising main 17 in. diameter built up of mild steel. From these tanks the water

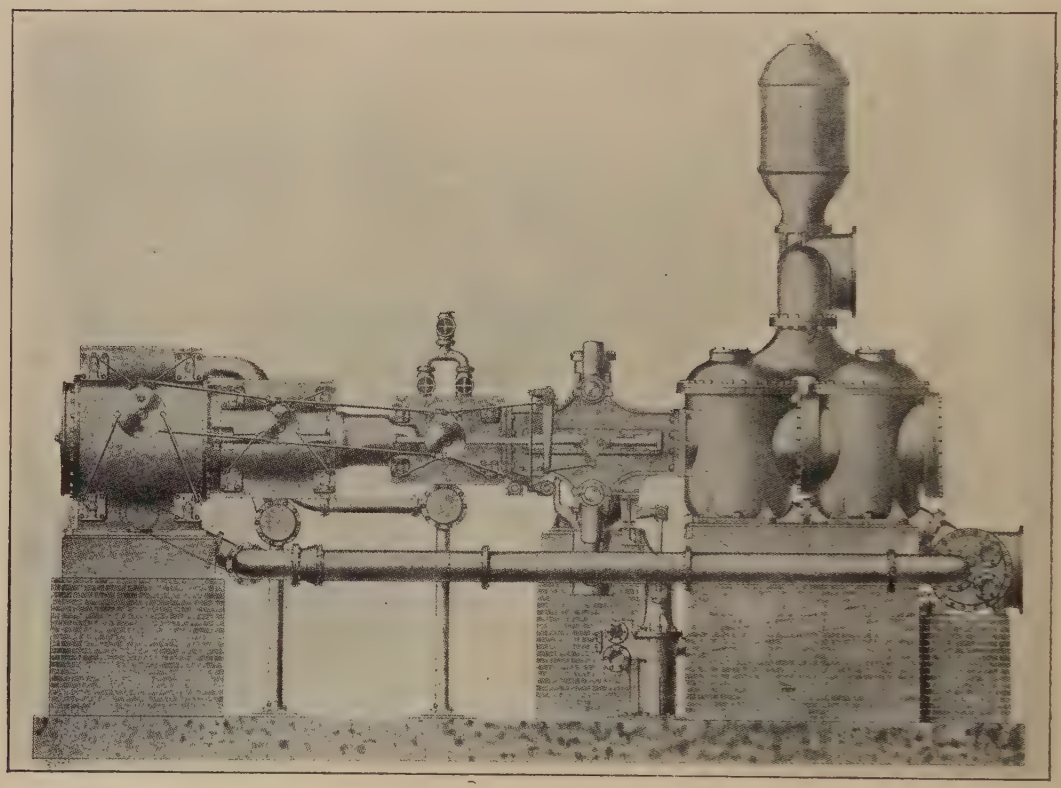


Fig. 92.—Triple-expansion Compensated Direct-acting Pumping Engine by Messrs. Worthington-Simpson, Ltd.

is delivered by the high lift force pump against a head of 465 ft. This pump is of the double-acting plunger type, 16½ in. diameter, the stroke of course being 5 ft. From a simple calculation it will be determined that the combined horse power of the pumps is 254, so that, assuming an engine efficiency of 90 per cent., the power developed by the main engine is about 280 i.h.p.

Attention has already been drawn to the engine and pumps shown in Plate III, but a few particulars as regards output, etc., may be given here. The unit was designed and constructed by Messrs. Ashton, Frost & Co., Ltd., and installed by them at the Rammey Marsh Pumping Station of the Metropolitan Water Board. The engine is of the horizontal cross compound surface condensing type, the high-pressure cylinder being 22 in. diameter and the low-pressure cylinder 34¾ in. diameter with a common piston stroke of 3 ft. The distribution of the steam is carried out by means of Corliss valve gear, the

cut-off of the steam to the high-pressure cylinder being controlled by the centrifugal governor mounted between the cylinders as shown. The pumps are of the double-acting high lift type, with rams driven direct from the tail rod of each cylinder, and there are no deep well pumps, the pumps drawing their supply from the suction tank immediately below the pumps in a similar manner to the Hammersmith pumps (see Fig. 88). The diameter of the rams of the high lift pumps, which are made of gun-metal with conical ends, is 10½ in., the stroke being, of course, 3 ft. The pumps are designed to deliver 4,500,000 gallons of water per day against a head of 360 ft. when the engine is running at 75 revs. per min. This is a very fine example of a modern high-powered pumping unit.

The horizontal engines described above are all of what we may term the single compound or triple-expansion type—that is to say, there is only one high-pressure cylinder, the steam from which exhausts into the intermediate and low-pressure cylinders before passing to the condenser. We have, however, already mentioned that for large units the " twin tandem " arrangement is often employed—that is to say, the engine consists of two high-pressure cylinders having their axes parallel, the steam from each exhausting into the intermediate and low-pressure cylinders, which are arranged in tandem with the high-pressure cylinders. Then from an extension of the piston rods the pumps are driven in the usual way either directly in the case of high lift pumps or through rocking levers in the case of deep well pumps. This type of pumping unit has been developed with considerable success by Messrs. Worthington-Simpson, Ltd., and has been installed by them in some of the largest pumping units, capacities up to 10,000,000 gallons per 24 hours having been obtained. The general arrangement of these "high duty" pumping units will be understood from Fig. 92, which shows the elevation of what the makers term their triple-expansion compensated direct-acting pumping engine. It will be noticed that the engine is not of the rotative type—that is, it is not fitted with a crank shaft and flywheel as in all the units already described; a compensating device is, however, fitted in order to maintain a practically uniform effort at the pump plungers throughout each stroke. It is claimed by the makers that as a result of this arrangement the economy of the engine is greater than that of the rotative type owing to the high mechanical efficiency, due, of course, to the absence of crank shaft and connecting rod bearings. Further, owing to the direct action of the effort from the steam cylinders on the pump plungers combined with the effect of the compensating device, a constant velocity of the water is developed in the discharge main from the pumps, and a considerable amount of power is saved as compared with the rotative engine, where if the flywheel has a uniform speed of rotation the pistons must have a continually varying velocity between zero at the beginning of the stroke, a maximum at the middle, and zero again at the end.

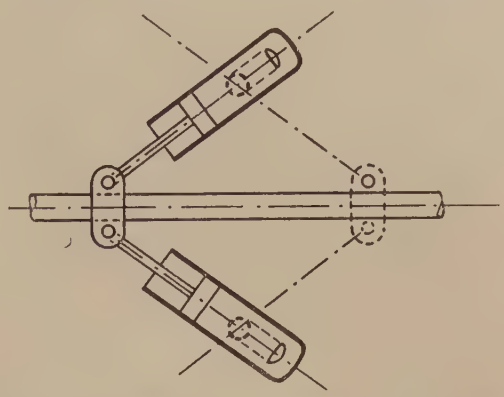


FIG. 93.—Diagrammatic arrangements of Compensating " Pots " for Direct-acting Pumping Engines.

It will be noticed from Fig. 92 that the engine is of the triple-expansion type with the cylinders arranged with their axes in the same straight line, the pump being driven from an extension of the high-pressure piston rod. Between the high-pressure cylinder and the pump is the compensating device which consists virtually of two small cylinders, or " pots " as they are termed, which are fitted with pistons or plungers worked off the

main pump rod as shown diagrammatically in Fig. 93. The compensating pots are supplied with water from an accumulator at a pressure of about 200 lb. per sq. in., so that during the first part of the stroke of the main engine the pressure in the pots exerts a retarding influence on the main pistons and energy is stored up in the accumulator. During the second half of the stroke, this energy is given out, assisting the expanding steam and maintaining more uniform conditions at the pump plungers. It will be observed from Fig. 92, that the valve gear on all three cylinders is of the Corliss type driven off the same crosshead as the compensating pots. The steam from the low-pressure cylinder passes to the surface condenser shown on the extreme right, which is placed in the suction pipe of the main pumps, the water to the pumps thus acting as circulating water to the condenser. The air pump is also driven off the same crosshead as the compensating pots, a bell crank lever serving to give a vertical motion to the air pump bucket. Owing to the fact that Fig. 93 shows a geometrical elevation of the unit, it is not possible to indicate whether it is of the "twin" type or not. These engines are indeed supplied in either one line or two lines according to the pump capacity required, but for the larger units the two-line arrangement is adopted. One of these high duty pumps is shown in Fig. 94, which is a reproduction of a photograph of one of two units built by Messrs. Worthington-Simpson for the Rangoon Waterworks, India. As will be seen from the photograph, the engines are of the twin tandem triple-expansion compensated type driving double-acting pumps which deliver the water through a breeches pipe to a common main as shown. The two engines are similar in every respect, the high-pressure cylinders being 16 in. diameter, the intermediate-pressure cylinders 26 in. diameter, and the low-pressure cylinders 50 in. diameter, the pistons all having a stroke of $26\frac{1}{2}$ in. The pump plungers are 48 in. diameter, and each unit is designed to deliver 14,400,000 gallons of water per day against a head of 200 ft. Thus the pump horse power is just over 600, and assuming an efficiency of 90 per cent., the engines develop about 670 horse power, each line being of course responsible for half this amount. The general features of the design of these engines are clearly shown in the photograph, and some idea of the overall dimensions will be obtained by comparison with the man in the foreground.

As already noted, these horizontal triple-expansion engines are also employed for the purpose of operating deep well pumps. As an example of this, a brief description may be given of the units supplied by Messrs. Worthington-Simpson to the municipal waterworks at Watford. The rapid growth of this pretty Hertfordshire town which followed the construction of the electric railway linking it with London necessitated the extension of the water supply. A new scheme was therefore prepared which would give an ample supply to meet both present and future needs, and it will be of interest to give a few of the main features of this scheme as indicating the nature of the complete installation. It having been proved that water in ample quantities was available, four pumping wells, each 36 in. diameter and 350 ft. deep, together with a 6 ft. diameter well 300 ft. deep, were sunk, the smaller wells being connected together by a large heading driven from the 6 ft. well. All the wells were lined with cast-iron tubes for a depth of 80 ft. to prevent the infiltration of the surface water. Pumping machinery was installed to draw water from the four 36 in. wells, the larger well being sunk chiefly in order to provide a working shaft from which to drive the headings to augment the supply should the smaller wells not yield sufficient for the requirements of the town. The pumping machinery consists of two sets of twin tandem triple expansion compensated engines, each set being equipped with a deep well pump and a high lift pump. One of these sets is sufficient to meet the present requirements of the town, so that the other is available as a stand-by.

The main engines are similar to those described above, the valves being of the Corliss type, the point of cut-off of the steam to each cylinder being adjustable by hand while

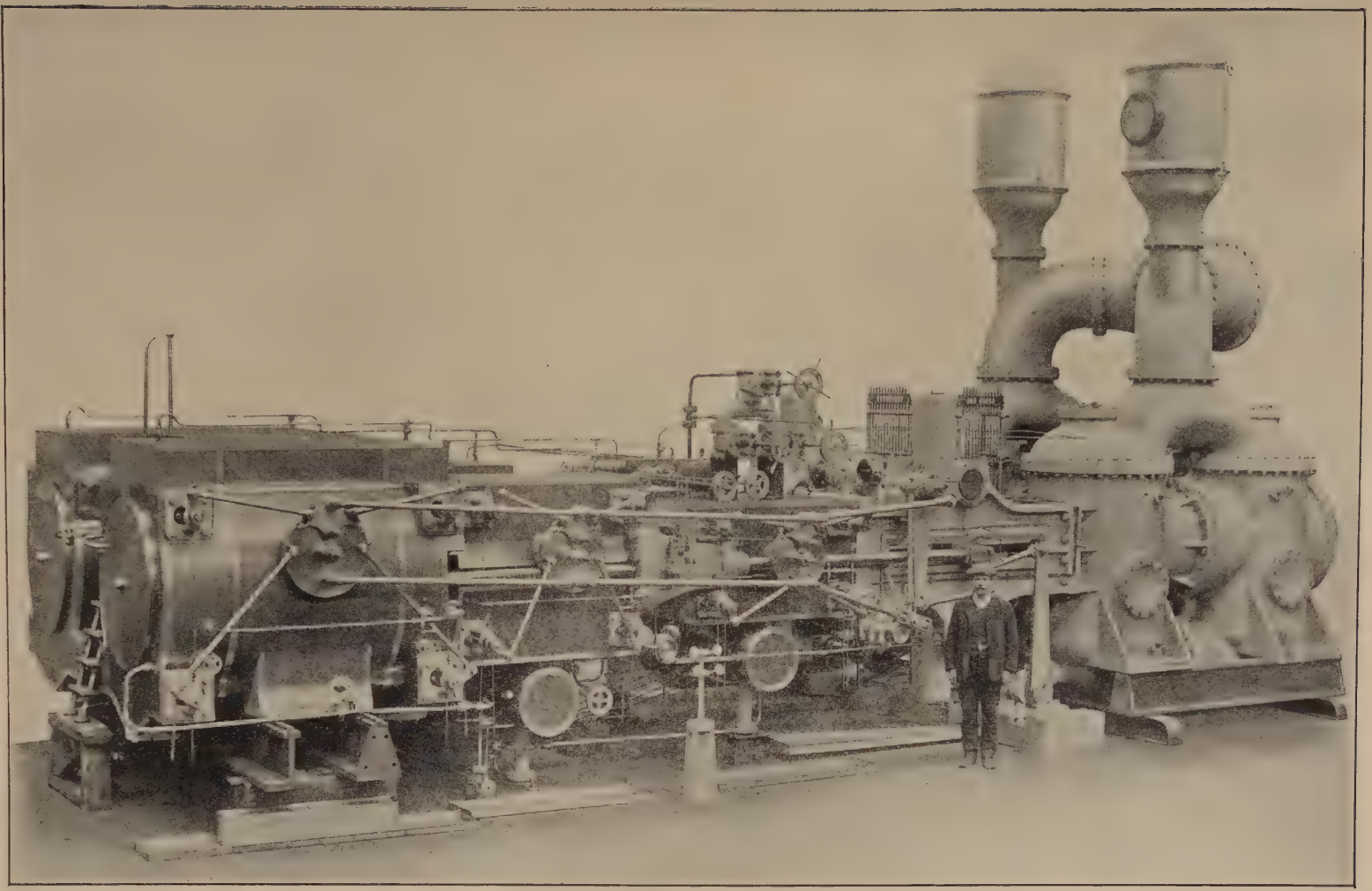


FIG. 94.—Twin Tandem Triple-expansion Compensating Pumping Engine, designed and constructed by Messrs. Worthington-Simpson, Ltd., for the Rangoon Waterworks, India. Capacity, 14,400,000 gallons per day against a head of 200 ft.

the engines are running. The general appearance of the complete set will be gathered from Fig. 95, which is a reproduction from a photograph taken in the engine room at Watford showing one of the sets as erected, the top of the low-pressure cylinder of the second set being seen on the left in the foreground of the picture. The high lift pump is immediately in front of the engine, with the deep well pumps beyond. These latter can only be just discerned in the picture. At present, the water is drawn from the bore wells at a depth of 150 ft. below the surface, at which level the desired quantity is obtained. Provision has, however, been made so that, should it be necessary, the water can be pumped from a depth of 350 ft., under which condition the steam pressure at the engines would be raised from the present working pressure of 160 lb. per sq. in. to 200 lb. per sq. in., for which pressure the boilers and engines are designed. The water from the deep well pump is delivered into a water-softening plant comprising a large covered-in settling tank built of reinforced concrete and having internal dimensions 80 ft. 2 in. by 44 ft. 3 in., with a depth of 9 ft. 9 in., giving a total capacity of 200,000 gallons. To the water in this tank a saturated solution of lime is added, precautions being taken to ensure that the lime is thoroughly mixed with the water. The "hardness" then settles out, the major portion of it being removed in the settling tank and the remainder in filters through which the water is passed before it gravitates to the suction tank in the engine room basement. From this tank it is drawn by the high lift pump through the surface condensers and delivered to the service reservoirs against a head of 320 ft. With one engine running, water is delivered from the deep well pumps at the rate of 2,500 gallons per minute, or 1,500,000 gallons per working day of 10 hours, the high lift pumps being capable of dealing with the same quantity of water. Thus with the deep well pump drawing water from a depth of 150 ft. the total lift is 470 ft., and the pump horse power is 356, while at the maximum depth of 350 ft. the pump horse power is about 500. Superheated steam is provided by two Lancashire boilers each 8 ft. 6 in. diameter by 30 ft. long, fitted with mechanical stokers, while a Green's economiser is installed for heating the feed water before it is pumped into the boiler. This plant may be regarded as thoroughly representative of modern practice in the design and equipment of an efficient municipal water supply scheme, although it will be recognised that the actual details of the plant must always depend upon local circumstances.

In the foregoing paragraphs we have described a number of different types of pumps such as are to be found in the various waterworks and sewage pumping stations that are to be found in almost every large town. But while the capacity of these pumps may be considerable, and while, indeed, some of them may be employed in raising the water against considerable heads, or, in other words, giving high lifts, they can all be classified as "low-pressure" pumps. A few figures will serve to verify this statement. Suppose we consider a unit in which the high lift pump delivers water against a head of 350 ft. We have already seen that the relationship between the head and the pressure is given by the simple formula $p = wh$, where p = pressure in lb. per sq. ft., w = weight per cu. ft. of water, and h = head in ft. Thus, since for water $w = 62{\cdot}5$, when $h = 350$, $p = 62{\cdot}5 \times 350 = 2{,}1875$ lb. per sq. ft., or, using the more familiar unit for pressure, $\frac{2{,}1875}{144} = 152$ lb. per sq. in. Similarly a deep well pump giving a lift of 175 ft. would be working against a pressure of 76 lb. per sq. in. These pressures may be taken as representing average practice in this particular branch of pumping work. When, however, we come to consider hydraulic power engineering in which water under pressure is utilised for operating presses, hoists, etc., it will be found that the working pressures are many times greater than those given above and that they can indeed be regarded as high pressures.

The average pressure employed in most hydraulic power installations is somewhere

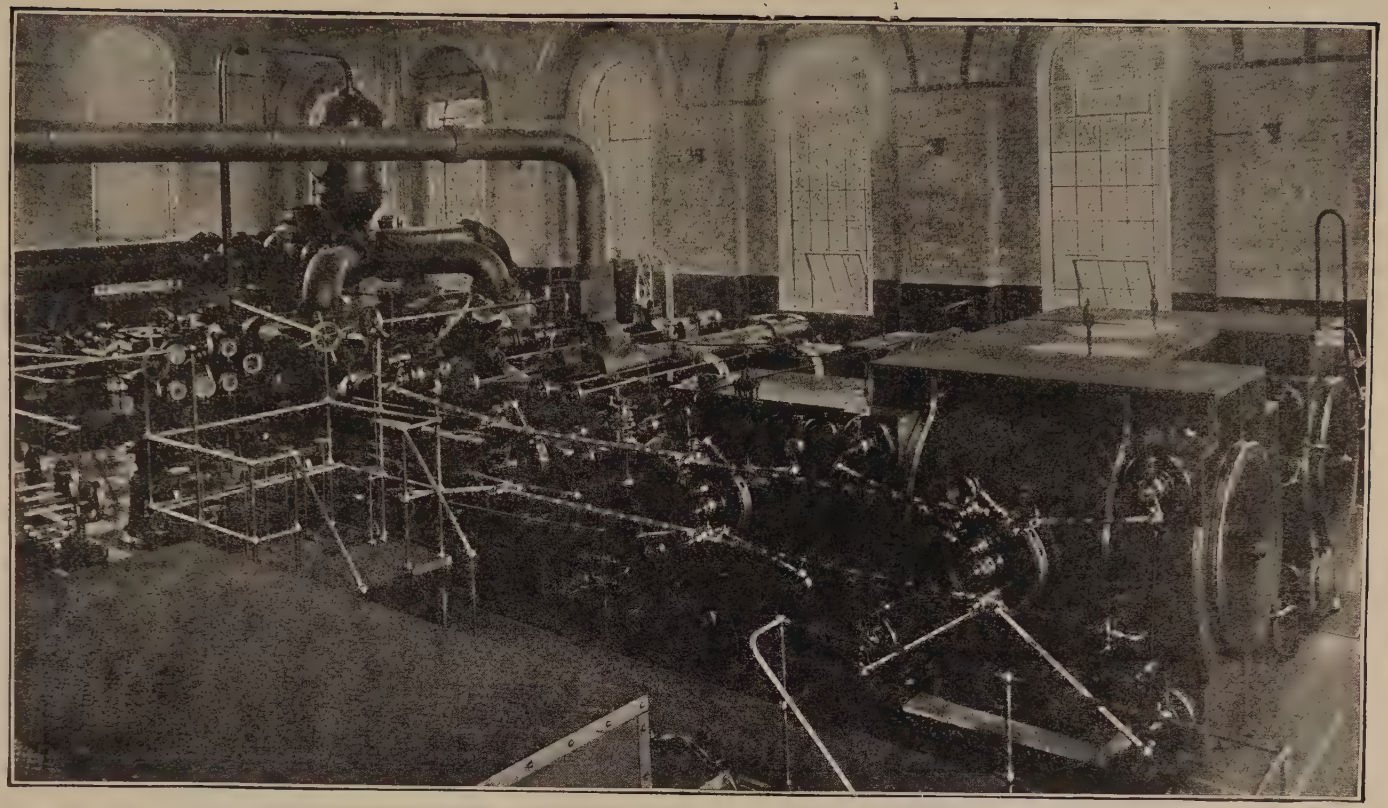


FIG. 95.—Worthington-Simpson Twin Tandem Triple-expansion Engine of 500 h.p., driving Deep Well and High Lift Pumps at the Municipal Waterworks, Watford.

in the region of 1,000 lb. per sq. in., although modern practice appears to favour far higher pressures than this, pressures up to 4 or 5 tons per sq. in. being sometimes used. Considering, however, the average of 1,000 lb. per sq. in., a simple calculation will show that a pump producing such a pressure as this is working under a head of 2,300 ft., while for a working pressure of 5 tons per sq. in. the head would be 25,800 ft. Of course it is not usual to speak of the head of water in connection with hydraulic installations, but these figures are given so that a comparison can be made between the duty of the hydraulic pump and the waterworks pump. On the other hand, the quantity of water delivered by the hydraulic pump is considerably less than that delivered by the low-pressure pump. Even assuming that the power required to drive the pump were the same, it is obvious that if the head is ten times as great, only one-tenth of the weight of water will be lifted. Actually, however, except in the largest installations, the power required in hydraulic installations is small. The great advantage of hydraulic power lies in the very large forces which can be obtained as a result of the water at high pressure acting on a piston. Thus, if we have a hydraulic press in which the plunger is 12 in. diameter and the working pressure of the water is 1,000 lb. per sq. in., the force on the plunger is = pressure × area = $1{,}000 \times \frac{\pi \cdot 12^2}{4} = 113{,}000$ lb. = 50·6 tons. It is obvious, therefore, that the chief value of hydraulic machinery lies in the comparatively simple method whereby enormous forces can be obtained. A further great advantage of hydraulic power lies in the fact that it is readily possible to store the high-pressure water, and thus the size of the pumping plant need only be sufficient for the average requirements. This is of considerable value, particularly where hydraulic presses are employed. These machines utilise a considerable amount of power during the time the actual press takes place, but a long interval of time may elapse between one press and the next, as, for example, when the end plates of boilers are being shaped and flanged. Thus it is of great advantage to be able to draw the high-pressure water from storage rather than that it should be generated by the pump as fast as it is required. The water at high pressure is stored in what is termed an accumulator, and one of these will be found on almost all hydraulic pipe lines. The hydraulic accumulator consists essentially of a large cylinder fitted with a heavily weighted plunger or ram. This plunger is lifted by the pressure of the water in the hydraulic main, and water at high pressure is stored in the accumulator. When, however, the pressure in the main has a tendency to fall, due to an excessive demand for water by the various machines, the plunger gradually falls and water is forced out from the accumulator into the main. A simple form of accumulator is shown diagrammatically in Fig. 96, this form being known as the weight case type, since the load on the plunger is due to a case built up of steel plates which is loaded with any heavy material, such as scrap iron, iron ore, or stones, until the desired pressure is obtained. In some cases the top of the ram carries a beam from which a number of cast-iron discs are slung. Obviously, the weight on the ram divided by its area gives the working pressure in the hydraulic main. The weights on the accumulator are usually fitted with a tappet arrangement in connection with the engine stop valve, so that when the accumulator is full, i.e. when the ram is at the top of its stroke, the steam supply to the engine is entirely cut off. As

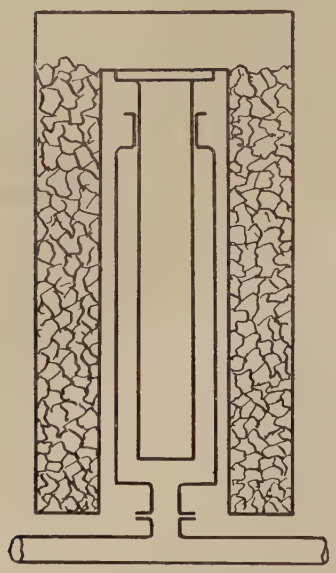


Fig. 96.—Diagrammatic Arrangement of Hydraulic Accumulator.

the weight falls due to the use of the water in the hydraulic machinery, the throttle is gradually opened, the engine driving the pumps and supplying water at the working pressure to the main. In this way the accumulator is kept nearly full except when a large and sudden demand is made upon it, and then the engine begins to operate at its full power to replenish the accumulator. It will be apparent that the use of the accumulator leads to a considerable reduction in the size of pumping unit which has to be installed in conjunction with a given amount of hydraulic machinery, the actual size depending upon the nature of the machinery being driven. In the same way the capacity of the accumulator will depend upon the extent of the sudden demand which is likely to be made by the machinery. In any form of machine using hydraulic power it will be found that the distance through which the plunger moves is comparatively small, and thus the actual amount of work done is small, since work done = force × distance, and as the horse power is measured by $\frac{\text{work done per min.}}{33{,}000}$, it follows that the actual horse power required to develop the high pressure in the water used for operating hydraulic machinery is also comparatively small. This will be at once apparent if we describe a few typical forms of hydraulic pumps. By this we mean pumps in which the pressure of the water used for operating hydraulic machinery is raised from atmospheric to its working pressure, and not, as is sometimes meant, pumps which are themselves driven by hydraulic motors.

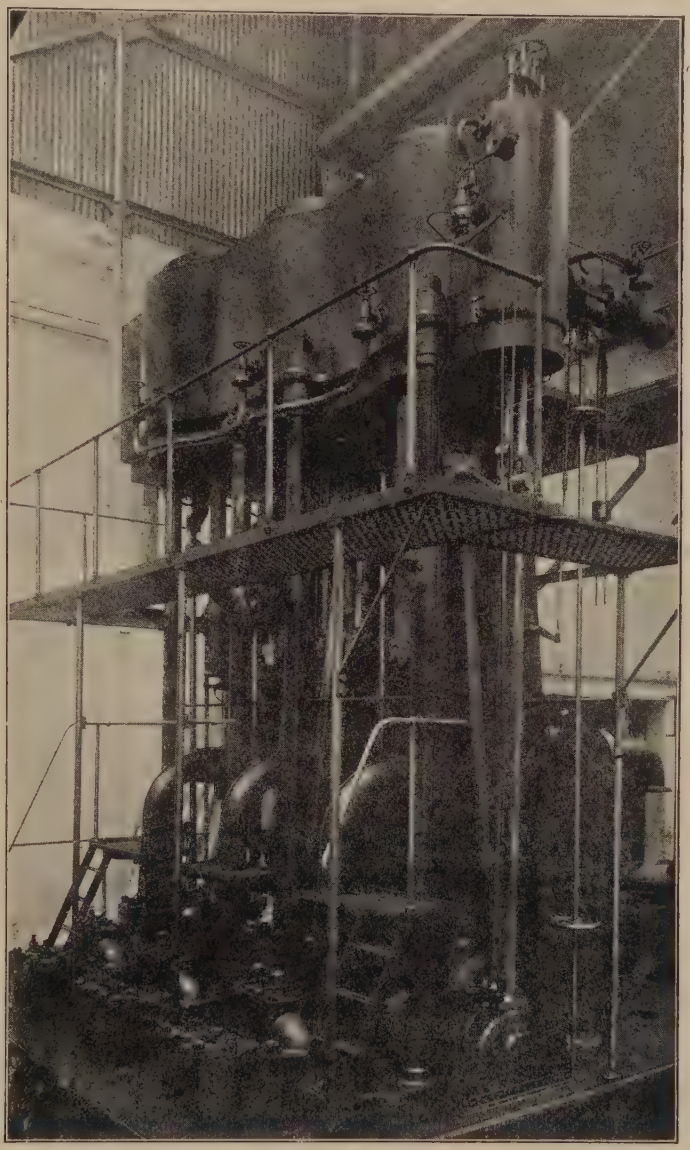


FIG. 97.—Vertical Triple-expansion Engine driving Hydraulic Pumps, delivering water at a pressure of 850 lb. per sq. in. Designed and Constructed by the Hydraulic Engineering Co., Ltd., Chester.

For hydraulic power stations where a large quantity of water at high pressure is generated, the vertical triple-expansion engine is very often employed for operating the

pumps. As an example of such a unit, attention may be drawn to Fig. 97, which shows a set designed and constructed by the Hydraulic Engineering Co., Ltd., of Chester, a firm which specialises in high-pressure pumps and hydraulic machinery. The engine is of the inverted vertical triple-expansion type the design of which follows marine practice.[1] The cylinders are supported at the back by cast-iron columns which serve to carry the back crosshead guides, and at the front by steel columns to which the front crosshead guides are secured. Following marine practice, the condenser is arranged at the back integrally with the engine, while the air and circulating pumps are driven through working levers from one of the engine crossheads, in this case that in the intermediate pressure line. The front and back columns are carried in a cast-iron bedplate which carries the crank shaft, upon which is a flywheel, the engine being of the rotative type. The pumps, of which there are three, are bolted vertically to facings on the front side of the condenser, the piston rods of the steam cylinders being coupled direct to the pump plungers, which thus have the same stroke as the engine pistons. The diameter of the plunger is, however, comparatively small, and thus the volume of water delivered is, as noted above, also small. The suction and delivery valves are placed in front of the engine at the level of the bedplate, and are connected to the pumps by means of curved steel pipes as shown. The suction to the pumps is supplied from overhead tanks, and the delivery to the mains is at a pressure of about 850 lb. per sq. in.

A very good idea of the performance of this type of pumping engine will be gathered from the following summary of the trials carried out by the Hydraulic Engineering Co., Ltd., on a set recently installed by them. The engine was of the triple expansion type, having cylinders 15 in., 24 in., and 39 in. diameter with a piston stroke of 27 in. The pump plungers were each $5\frac{3}{4}$ in. diameter, and the engine was designed to deliver water to an accumulator at a pressure of 800 lb. per sq. in. The following results were obtained:

TRIAL OF VERTICAL TRIPLE-EXPANSION PUMPING ENGINE BY THE HYDRAULIC ENGINEERING CO., LTD., CHESTER

Duration of trial	8 hours
Average speed of engine, by counter	63·38 revs. per min.
Average steam pressure at engines	183·3 lb. per sq. in.
Average temperature of superheated steam	478·7° F.
Average vacuum in condenser	28·9 in.
Barometric pressure	29·74 in.
Pressure in accumulator	800 lb. per sq. in.
Water delivered to accumulator	28,785·6 gallons per hour
Water delivered to accumulator	479·76 gallons per min.
Pump horse power in main pumps	267·51
Average mean pressure in h.p. cylinder	60·16 lb. per sq. in.
Average mean pressure in i.p. cylinder	27·22 lb. per sq. in.
Average mean pressure in l.p. cylinder	11·58 lb. per sq. in.
Indicated horse power, h.p. cylinder	88·98
Indicated horse power, i.p. cylinder	104·75
Indicated horse power, l.p. cylinder	118·89
Total indicated horse power	312·62
Mechanical efficiency $= \frac{\text{pump h.p.}}{\text{indicated h.p.}}$	85·57 per cent.
Total feed water to boilers	31,860 lb.
Feed water per hour	3,983 lb.
Feed water per pump h.p. per hour	14·88 lb.
Feed water per indicated h.p. per hour	12·74 lb.
Jacket water per hour	295 lb.
Percentage of jacket water to feed	7·42 per cent.
Water collected from glands (not included in the above)	10 gallons

[1] See Chapter V of this volume for a full description of a triple-expansion marine engine.

The performance of the engine as expressed by the figure 12·74 lb. of feed water per i.h.p. per hour, i.e. 12·74 lb. of steam per i.h.p. per hour, is very satisfactory for an engine of this type, while the mechanical efficiency, which includes, of course, the losses in the pumps as well as in the engine, is also very good. It may be noted that the figure for the pump horse power can be verified after converting the pressure in the accumulator to a head.

Thus: 800 lb. per sq. in. $= \frac{800 \times 144}{62{\cdot}5} = 1{,}844$ ft. head, and

$$\text{Pump horse power} = \frac{1{,}844 \times \text{weight of water delivered per min.}}{33{,}000}$$

$$= \frac{1{,}844 \times 479{\cdot}76 \times 10}{33{,}000} = 267{\cdot}5.$$

Another type of engine which is sometimes employed for driving hydraulic pumps is the twin horizontal compound engine. It would of course be possible for small

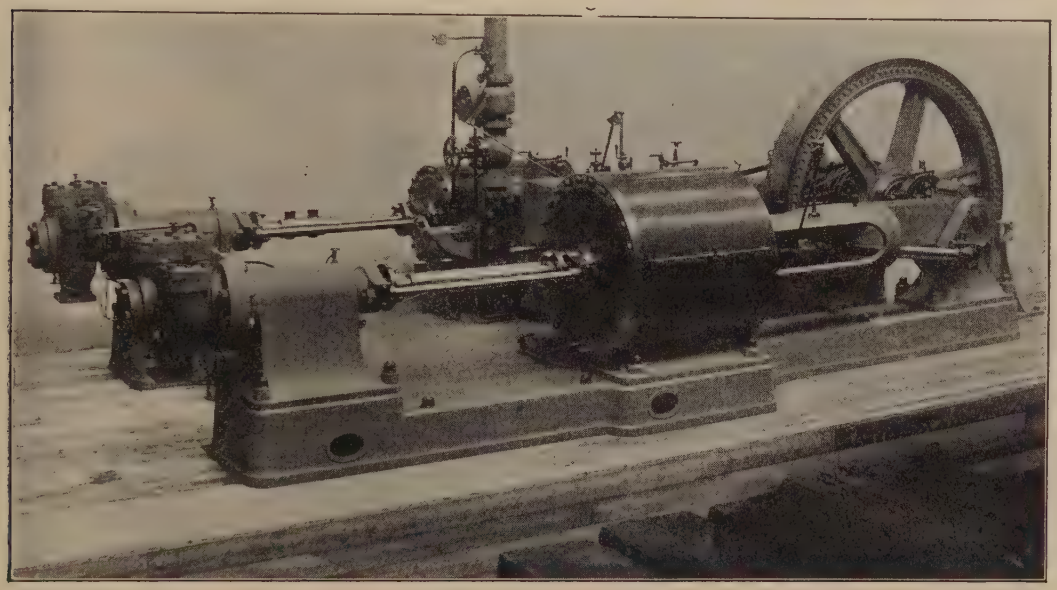


Fig. 98.—Twin Horizontal Compound Hydraulic Pumping Engine by the Hydraulic Engineering Co., Ltd.

installations to employ a single-cylinder horizontal engine, but the twin arrangement—sometimes with two high-pressure cylinders instead of a high-pressure and a low-pressure cylinder—gives a better balanced engine and one which, having its two cranks at right-angles to each other, can start in any position. This is a particularly valuable feature when the engine stop valve is connected to the accumulator and automatic regulation is provided in order to keep the accumulator full. The general features of one of these horizontal compound pumping engines will be gathered from a study of Fig. 98, which shows a unit designed and constructed by the Hydraulic Engineering Co., Ltd. It will be noticed that the pumps are driven from extensions of the piston rod, and attention may be drawn to the small diameter of the pump cylinders relatively to their length. The pump cylinders were not visible in our view of the vertical triple-expansion engine, otherwise the same characteristic could have been observed in them also. This is of course due to the fact that the actual volume of water delivered per stroke is small. The steam cylinders, the pumps, and the crank shaft bearings are all bolted to a common base plate, the result being an engine of extreme rigidity, and owing to the wide area over

which the weight is distributed, the foundations can be made comparatively light. This is the great advantage of the horizontal type of unit. On the other hand, the floor space which it occupies is considerable.

For smaller outputs, or where it is preferred to employ an electric motor in preference to a steam engine, the electrically driven hydraulic pump is available. In order to obtain uniform delivery from the pumps, combined with an even turning moment on the crank shaft, three pumps arranged side by side are employed, these being driven from cranks arranged at 120 degrees to each other. A cast-iron bedplate is provided which carries the pumps, the crosshead guides, and the crank shaft together with the motor and its gearing, the whole forming a self-contained unit. The general appearance of one of these pumps as designed and constructed by the Hydraulic Engineering Co., Ltd., will be gathered from Fig. 99. In order to obtain a suitable speed at the crank shaft, it is necessary to employ gearing as shown, the ratio of reduction being sometimes as high as 10 to 1. If the power of the motor exceeds 50 b.h.p., it is desirable to employ an outer bearing to support the pinion in order to relieve the armature shaft of undue bending stress, this arrangement being adopted in the unit shown. Below 50 h.p. the outer bearing can be dispensed with, the pinion being carried on an extension of the motor shaft. It is found that the cylindrical crosshead guide with circular crosshead gives the best results, as in addition to yielding a greater bearing surface, it ensures greater accuracy in alignment, the guides and the cradles for the pumps being machined in one setting. One disadvantage of the electrically driven pump is to be noted, viz. that it tends to accelerate very quickly. This rapid acceleration is a feature of the electric motor, but in conjunction with hydraulic pumps it tends to cause too rapid a rise of pressure in the hydraulic mains. To guard against the evil effects of this, the pump must be fitted with a relief valve and the motor with a circuit breaker, although it should be pointed out that these should seldom come into operation with properly designed plant. Some idea of the nature of the performance of these pumps will be gathered from the summary given on the following page of the trial of two sets of pumps of the type shown in Fig. 99, the specified duty of each pump being the delivery of 280 gallons of water per min. at a pressure of 750 lb. per sq. in. when the driving shaft is running at 60 revs. per min. The pumps are of the three-throw single-acting plunger type each 7 in. diameter with a stroke of 12 in. It is to be noted that during the trial the average head of water over the top of the seat of the suction valve was 4 ft. 3 in.

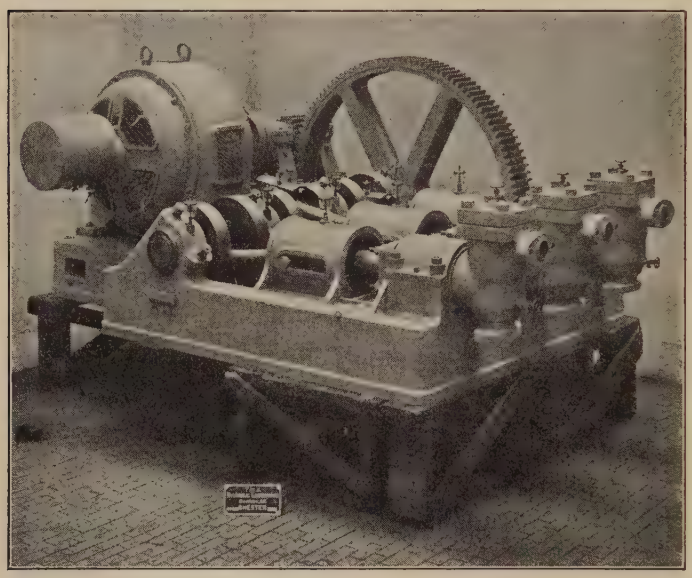


FIG. 99.—Electrically Driven Three-throw Hydraulic Pump, delivering 280 gallons of water per min. at a pressure of 750 lb. per sq. in.

Mention has already been made of the fact that working pressures far higher than the average of 800 lb. per sq. in. are frequently employed. The Hydraulic Engineering Co.

TRIALS OF TWO SETS OF HORIZONTAL ELECTRICALLY DRIVEN THREE-THROW PLUNGER PUMPS BY THE HYDRAULIC ENGINEERING CO., LTD., CHESTER

	Pump No. I.	Pump No. II.
Duration of trial, minutes	30	30
Average voltage at pumping station switchboard	459	465
Average ampères at pumping station switchboard	322	308
Average hydraulic pressure by gauge	725	715
Average revs. per min. of pump crank shaft	61·6	61·0
Average output from pumps, gallons of water per min.	297·6	296·3
Efficiency of pump volumetric,[1] per cent.	97	97
Electrical units used, in B.O.T.s by meter	68	65
B.H.P. calculated from meter readings	182	174
B.H.P. calculated from water pumped	152	151
Overall efficiency, per cent.	83·5	86·5

recently completed an order for two electrically driven three-throw pumps having a working pressure of 3,240 lb. per sq. in., and designed to give 1,200 gallons of water per min. The plungers were each 2 in. diameter with a stroke of 10 in., and the speed of the pump crank shaft was about 60 revs. per min. The general features of the design of the pump were exactly the same as for that illustrated in Fig. 99, with the various parts made heavier in order to withstand the higher working pressure. The following is a summary of the results obtained during the trials of these pumps.

TRIALS OF TWO SETS OF ELECTRICALLY DRIVEN PUMPS DELIVERING WATER AT 3,240 LB. PER SQ. IN. PRESSURE

	Pump No. I.	Pump No. II.
Speed of pump crank shaft, revs. per min.	61·3	60·0
Gallons of water per min.	19·9	19·7
Gallons of water per rev.	·325	·329
Theoretical displacement per rev.	·335	·335
Slip, per cent.	3	2
Working pressure, lb. per sq. in.	3,240	3,240
Ampères to motor (average)	205	205
Voltage (average)	212	207
Electrical horse power	58·3	56·9
Brake horse power of motor[2]	51·25	50·0
Pump horse power	45·0	44·7
Efficiency of pump, $\frac{\text{p.h.p.}}{\text{b.h.p.}}$, per cent	88	89·5
Overall efficiency, $\frac{\text{p.h.p.}}{\text{e.h.p.}}$, per cent.	77·1	78·6

For smaller outputs, or where the hydraulic power is only used at intervals, the belt-driven pump as shown in Fig. 100 is available. This pump, which is also by the Hydraulic Engineering Co., Ltd., is of the three-throw type, and as far as the pump itself is concerned, is substantially the same as the electrically driven unit described above. A gear wheel is mounted on the crank shaft, and this is driven by a pinion carried at the end of the pulley wheel shaft, this shaft being arranged over the pump guides as shown, thus making a very compact and serviceable unit. The shaft carries a fast and loose pulley

[1] The volumetric efficiency of a pump is the ratio of the volume of water actually delivered to the volume swept through by the plungers, the latter being N × A × L, where N = total number of pump strokes, A = area of plunger, L = length of plunger. It measures the amount of slip which occurs past the plungers, loss through leakage, etc.

[2] The efficiency of the motor was taken at 88 per cent.

at its other end, so that it can be driven from a line shaft. In this way the motor is being used for driving other machinery and is not standing idle when the pump is not required. A further advantage of the belt drive lies in the fact that the vibrations from the pumps are not transmitted to the motor through the belt, and thus the wear and tear on the motor is considerably reduced.

In all that has so far been said in this chapter we have been considering the reciprocating pump, i.e. the pump in which the piston or plunger or bucket, as it is variously called, has a movement backward and forward in the pump cylinder. But just as the rotary steam engine or turbine has been developed in recent years and has proved in many ways superior to the reciprocating steam engine, so, too, the rotary pump is gradually becoming a very serious competitor of the reciprocating pump. The analogy between both types of engines and pumps is complete, since in the case of the engine, whether rotary or reciprocating, a fluid under pressure is supplied to the engine and mechanical work is obtained from it, while in both types of pump the reverse happens, mechanical work being supplied to the pump, and a fluid—usually water—under pressure is obtained from it. As, however, the action of the rotary pump depends upon centrifugal force, as a class these pumps are generally termed "centrifugal pumps," although the terms "rotary pump" and "turbine pump" are occasionally used. Centrifugal pumps are available for almost all kinds of pumping work, from delivering large quantities of water at very low head, as, for example, when circulating the water through a condenser, to supplying water for hydraulic machinery where the delivery pressure may exceed 1,000 lb. per sq. in. The development of the steam turbine and also of the electric motor, both of which are particularly suited to driving centrifugal pumps, has been largely responsible for the enormous increase in the use of pumps of this type which has occurred in recent years. We have already referred in passing to many of the applications of rotary pumps in connection with steam engine auxiliaries,[1] although except for the rotary feed pump, the head at which the water is delivered by these pumps is comparatively low. Now, however, we must consider this type in greater detail.

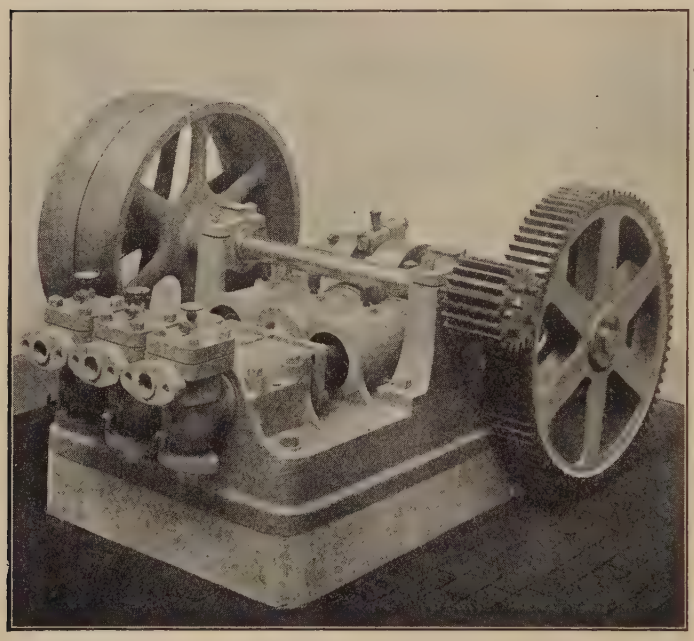


Fig. 100.—Belt-driven Three-throw Hydraulic Pump.

Before describing some of the various forms of centrifugal pump which are used in modern engineering practice, we must first of all briefly consider the principle underlying the action of these pumps. As already noted, it is centrifugal force acting upon water which is given a motion of rotation that is responsible for the delivery of a fluid from a centrifugal pump. Imagine an elementary form of pump such as is shown in Fig. 101

[1] See Volume I, Chapter VIII.

consisting of a number of flat vanes mounted on a shaft, the whole being carried in a casing provided with a suction and a delivery pipe as shown. If now, after the pump has been flooded, the vanes are rotated at high speed, the water will be whirled round in a circle, and in consequence it will all try to move outwards under the action of centrifugal force. Thus some of the water will find its way out of the delivery pipe and, provided the suction pipe is, theoretically, not more than 34 ft. (the height of the water barometer) above the level of the supply, water will flow into the pump casing to prevent the formation of a vacuum, and thus so long as the vanes are kept rotating the pump will continue to deliver water. It is apparent, however, that this arrangement will be very inefficient. The water will be churned up in the casing, pressing against the outer surface, but only a small portion of it will find its way through the delivery pipe. Whenever water is in motion, being churned up or agitated in any way, mechanical energy is required to produce that motion; this energy is therefore not available for doing useful work in actually delivering water from the pump, and hence the efficiency is low. It is certain, therefore, that this type of pump would not raise water through a height of 34 ft. above the source, nor would the delivery head be very considerable. It illustrates, however, the fundamental principle of the centrifugal pump.

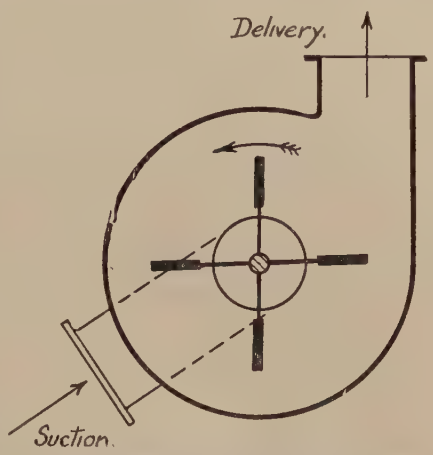


FIG. 101.—Diagrammatic Section of Simple Form of Centrifugal Pump.

In designing an efficient form of centrifugal pump, we must endeavour as far as possible to provide a smooth path for the water through the pump, so that at no point is it subjected to any sudden change of velocity. If this can be accomplished, a pump of high efficiency will be obtained. Suppose, therefore, instead of the pump shown in Fig. 101 we have an arrangement as shown in Fig. 102. As before, we have the vanes mounted on a shaft and rotated at high speed in a casing, but now the vanes are of a particular curved form carried between two side discs, while the water, instead of finding its way into the casing in a haphazard manner, is led to the centre of the wheel, whence it is compelled to move outwards, being delivered from the outer periphery into the casing, the side discs preventing the spilling of water from the sides of the vanes. Moreover, the wheel, or the impeller, as it is usually termed in centrifugal pump practice, is mounted eccentrically in the casing, so that as the water is delivered from the wheel it moves smoothly towards the delivery pipe, where it is finally discharged. Even this does not represent the most efficient form of centrifugal pump, but it is obvious that it is far more efficient than the primitive type described above. It will be observed that the vanes on the impeller are curved backward relatively to the direction of rotation of the wheel, although the student is often inclined to think that they should be curved the other way round (or that the direction of rotation should be in the opposite way to that shown), so that the vanes may, as it were, scoop up the water. A moment's consideration will show that this would not yield an efficient pump. It has already been noted that the water must pass through the pump without any sudden change of velocity, entering and leaving without shock, and hence the vanes must be so designed that the water moves smoothly through the wheel, which it would not do if the direction of rotation were reversed. It will be of interest to consider the exact path of the water through the wheel.

Consider a particle of water at A in Fig. 103, which shows one blade of the impeller, the moment before it enters the wheel, the pump being assumed to be running at normal

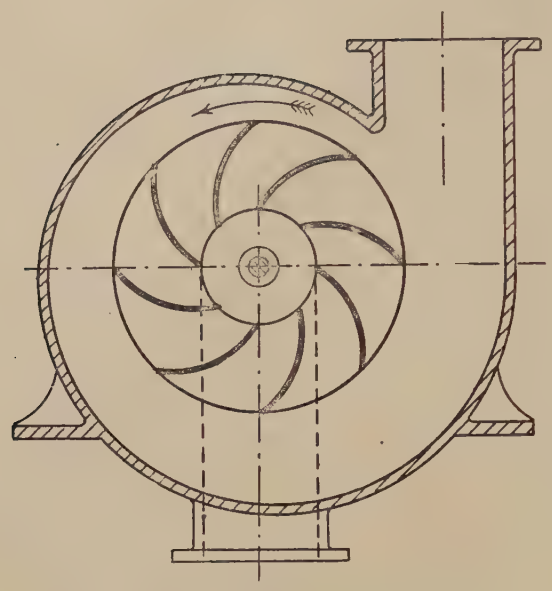


FIG. 102.—Sectional Elevation of Centrifugal Pump.

speed. The water has been drawn up the suction pipe and, following the path of the particle of water immediately before it, may be said to be moving radially outward with a velocity v_1, as shown. The tip of the blade is, however, moving with a velocity v_2 tangential to its circular path, and hence, by the well-known rule for relative velocity as explained in any elementary textbook on mechanics, the velocity of the particle of the water relative to the tip of the vane is given by v_3. Hence if the water is to enter the wheel without any change of velocity either in magnitude or direction, the vane must be inclined at such an angle that v_3 is tangential to the vane. Or, to look at it from another point of view, if the particle of water immediately on entering the wheel at A acquires the velocity of the vane tip v_2 and at the same time moves along the vane with a velocity v_3, its resultant velocity both in magnitude and direction will be given by v_1, which is the diagonal of the parallelogram between v_2 and v_3. Thus the velocity of the water is precisely the same before entering the wheel as it is after, and there is no shock as the water enters the wheel, moreover, v_1 is the direction of the velocity imparted to the water by the centrifugal force acting upon it. If now we imagine that the water slips over the surface of the vane, it will ultimately reach the point B where, immediately before leaving the vane, its velocity relatively to the vane may be considered as being V_3 tangential to the curve of the vane. The magnitude of V_3 will be greater than v_3 owing to the effect of centrifugal force on the water, the actual increase of velocity depending upon the size of the vane and its speed of rotation. But the outer tip of the vane B is moving round with the wheel, and thus it has a velocity V_2 tangential to the circular path traced by B, and V_2 is much greater than v_2, since the point B is at a greater distance than A from the centre of rotation of the wheel. Hence the resultant velocity of the water leaving the wheel will be represented both in magnitude and direction by V_1. Thus, as a result of passing through the wheel in which the vanes are of the form indicated by AB, the absolute velocity of the water is changed from v_1 to V_1, and instead of flowing radially, its direction is also changed as shown, so that it leaves the

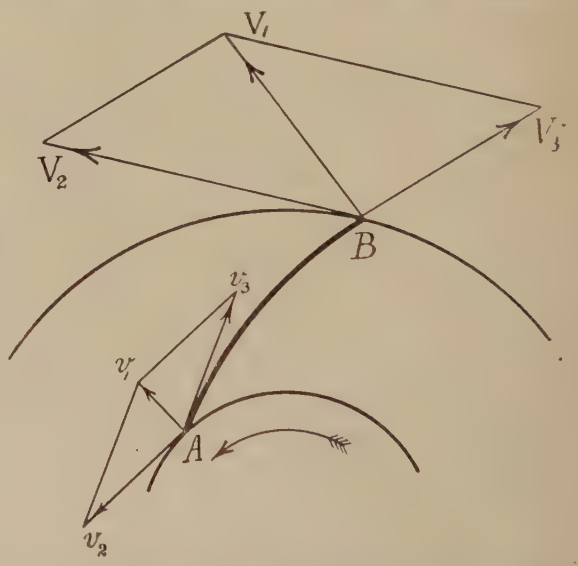


FIG. 103.—Diagram showing Velocity of Particle of Water passing through the Impeller of a Centrifugal Pump.

wheel in a direction more nearly tangential than radial. Thus it flows more or less easily through the space surrounding the wheel until it reaches the discharge pipe.

While, however, the space surrounding the wheel may be designed in order to enable the water to leave with the minimum of friction, it is found in practice that considerable loss of efficiency occurs owing to the high velocity of the water and the whirls and eddies that are set up in this space. Moreover, instead of a pump discharging large volumes of water at high velocity, it may be desirable to have a smaller discharge at a higher pressure, or, in other words, it may be desirable to discharge the water against a higher head. If, therefore, we can find ways and means for converting the velocity energy or kinetic energy of the water as it leaves the impeller into pressure energy, not only will this higher delivery head be obtained, but, owing to the reduction in the velocity, the loss of efficiency will be considerably less. It will be noticed in Fig. 102 that the space around the wheel gradually diverges, and thus the velocity of the water passing through it tends to become less and less—in other words, the pressure increases, since by the principle of the conservation of energy, a loss of energy of one kind must be accompanied by an increase of energy of another kind. The student will recall that in dealing with air extraction from steam condensers we saw that a diverging nozzle gave an increase of pressure in very much the same way.[1] The use of a diverging space, or volute casing as it is called, around the impeller yields a somewhat higher pressure at the delivery pipe owing to the decrease of velocity, but the increase of efficiency is very small owing to the eddies caused by the water discharged from the impeller into the volute casing. Nevertheless, owing to its simple form, this arrangement is adopted on large numbers of pumps where only a comparatively low head is required and where the efficiency of the pump is not of great consequence. In some types of pumps the volute-shaped casing is replaced by a volute or whirlpool chamber of circular shape which surrounds the impeller, the whirlpool chamber gradually increasing in diameter towards the discharge pipe. In others, a series of guide vanes are arranged in the space surrounding the impeller, these vanes forming a series of diverging spaces through which the water passes. This ring of guide vanes is sometimes referred to as the diffuser—precisely the same term as is used in connection with air ejectors for condensers—and the vanes are set at such an angle that the water enters them without shock. In order that this condition may be fulfilled, it will be realised that the guide vanes would be set so that V_1, the absolute velocity of the water leaving the impeller, in Fig. 103, is a tangent to the guide vanes at their inner circumference. The centrifugal

FIG. 104.—Centrifugal Condenser Circulating Pump by Messrs. Mather & Platt, Ltd. Installed at the Electricity Works of the Bolton Corporation. Capacity, 18,666 gallons per min. against a head of 28 ft.

[1] See Volume I. Chapter IX.

pump in which guide vanes are employed is sometimes called a turbine pump, owing to the fact that we have, as in the turbine, both fixed and moving blades. It certainly is one of the most efficient types of pump so far devised and one which is capable of discharging against a considerable head.

Where very high pressures are required, as for hydraulic work, it is usual to employ two or more centrifugal pumps mounted in series on the same shaft, each pump consisting of an impeller and a discharge chamber, the discharge chamber from the first pump leading from the periphery of the first impeller down to the centre of the casing, where it becomes the suction chamber for the second impeller, and so on. Each discharge chamber is made of diverging form, so that the kinetic energy of the water leaving the first impeller is partially converted into pressure energy before it enters the second

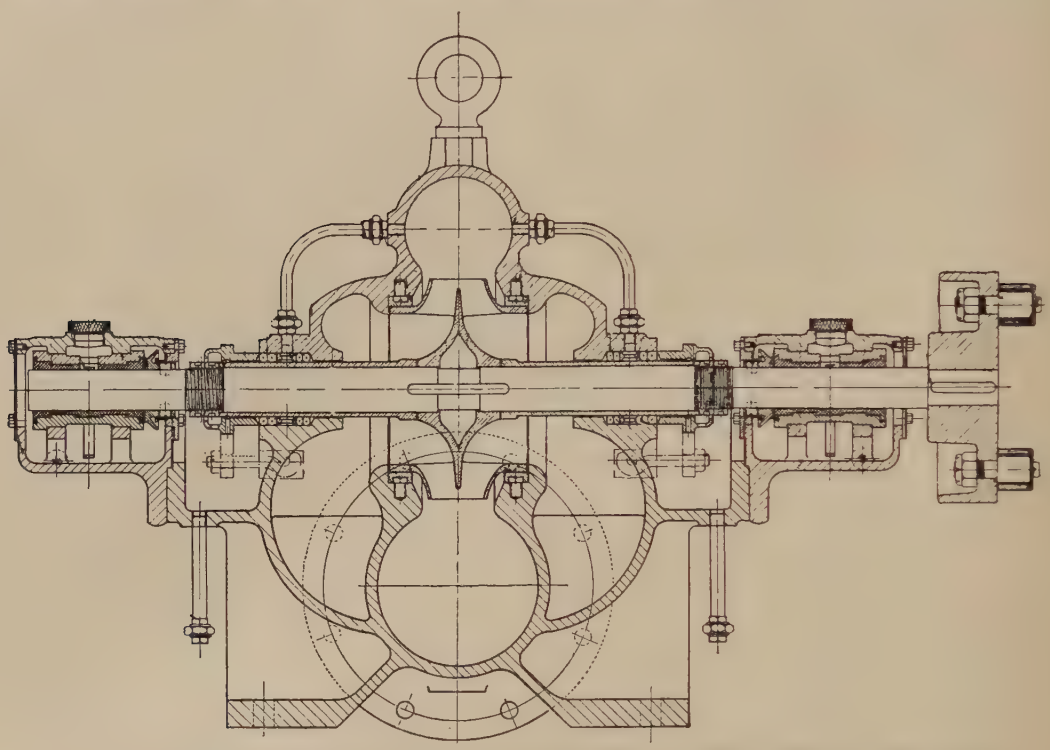


Fig. 105.—Sectional Elevation of Low Lift Centrifugal Pump by Messrs. Mather & Platt, Ltd.

impeller. Here its kinetic energy is increased without any appreciable change in pressure, until after leaving the second impeller this kinetic energy is again converted into pressure energy. Thus the pressure rises at each stage until the desired high pressure is obtained. It will be realised, of course, that in their practical details various makes of centrifugal pumps may differ to a considerable extent, but it will be found that the fundamental principles governing their action are the same in every case, and it is these principles which we have endeavoured to set forth, in very simple language, above. We must now describe one or two actual examples of the various types of centrifugal pumps.

For a number of years, the well-known firm of Mather & Platt, of Manchester, have made a speciality of the design and construction of centrifugal pumps for all kinds of service, and by the courtesy of Messrs. Mather & Platt, Ltd., we are enabled to include the following descriptions and illustrations of their pumps. Considering first of all pumps of low head, a number of standard sizes of varying capacity and lifts have been designed

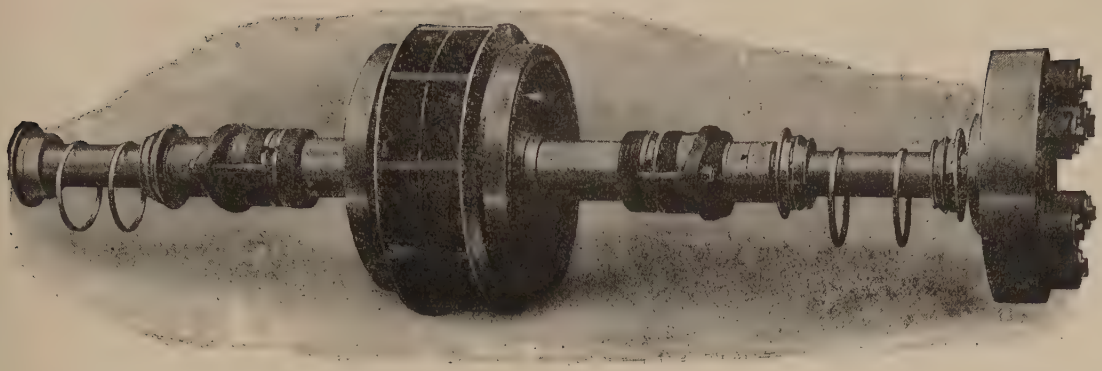


FIG. 106.—Shaft and Impeller for Low Lift Centrifugal Pump with Glands and Flexible Coupling.

for use in such service as circulating condensers, pumping out docks, irrigation work, etc., where the head against which the pump works does not exceed, say, 40 ft. The general appearance of these pumps will be gathered from Fig. 104, which shows one of two pumps supplied to the Electricity Department of the Bolton Corporation for circulating the cooling water through condensers. These pumps, which have impellers 30 in. in diameter, were designed to deliver 18,666 gallons of water per min. against a head of 28 ft. at a speed of 325 revs. per min., under which conditions the power required to drive the pump is 220 b.h.p. The principal features of the design will be gathered from a study of the sectional elevation drawing reproduced in Fig. 105, which, however, shows a rather smaller size of pump than that shown in Fig. 104. It will be noticed that the impeller is of the double type, water being admitted on both sides, thus avoiding the possibility of end thrust on the shaft. After leaving the impeller, the water is discharged into a chamber of spiral or volute form of gradually increasing diameter, until finally it is discharged from the pump at the delivery pipe. The casing is made of hard, close-grained cast iron split on the horizontal centre line to facilitate the inspection and removal of the impeller. As will be seen from Fig. 104, both the suction and the discharge branches are cast on the lower half of the casing, so that the top half can be lifted off without disturbing the pipes in any way. The impeller is of phosphor bronze, unless the liquid to be pumped is such that this metal

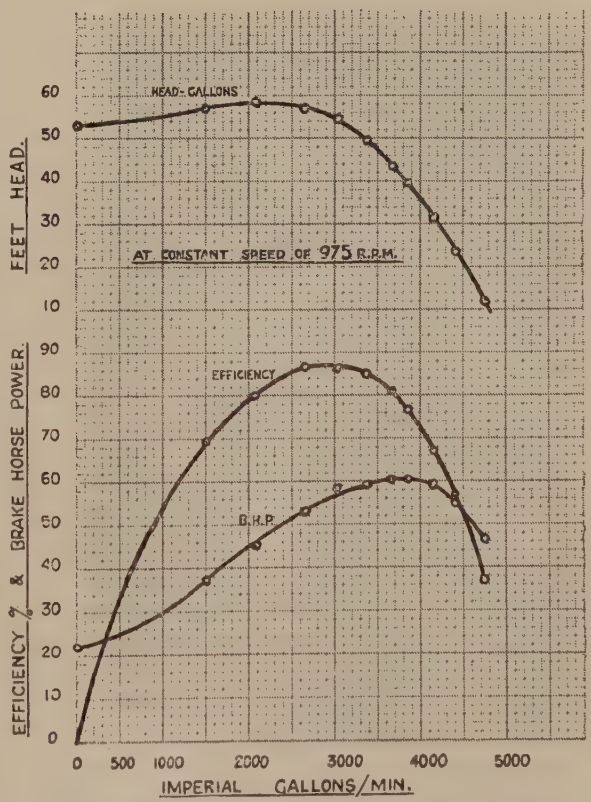


FIG. 107.—Characteristic Curves for a Mather & Platt Low Lift Centrifugal Pump.

cannot be used. It is turned and bored to gauge, and is carefully balanced to ensure smooth running. The vanes are filed to template, and the water passages are finished with a smooth surface in order to reduce the loss due to friction to a minimum.

The shaft to which the impeller is keyed is made of mild steel ground accurately to size, and provided with renewable phosphor bronze sleeves where it passes through the pump casing. In order to relieve the casing of all strains due to the weight of the impeller, glands are provided where the shaft passes through the casing, and the main bearings which support the shaft are arranged outside and quite clear of the casing. To prevent the leakage of water through the glands, different makers employ different forms of packing, but Messrs. Mather & Platt have found, as a result of many years' experience, that fine cotton treated with tallow and graphite forms the best kind of

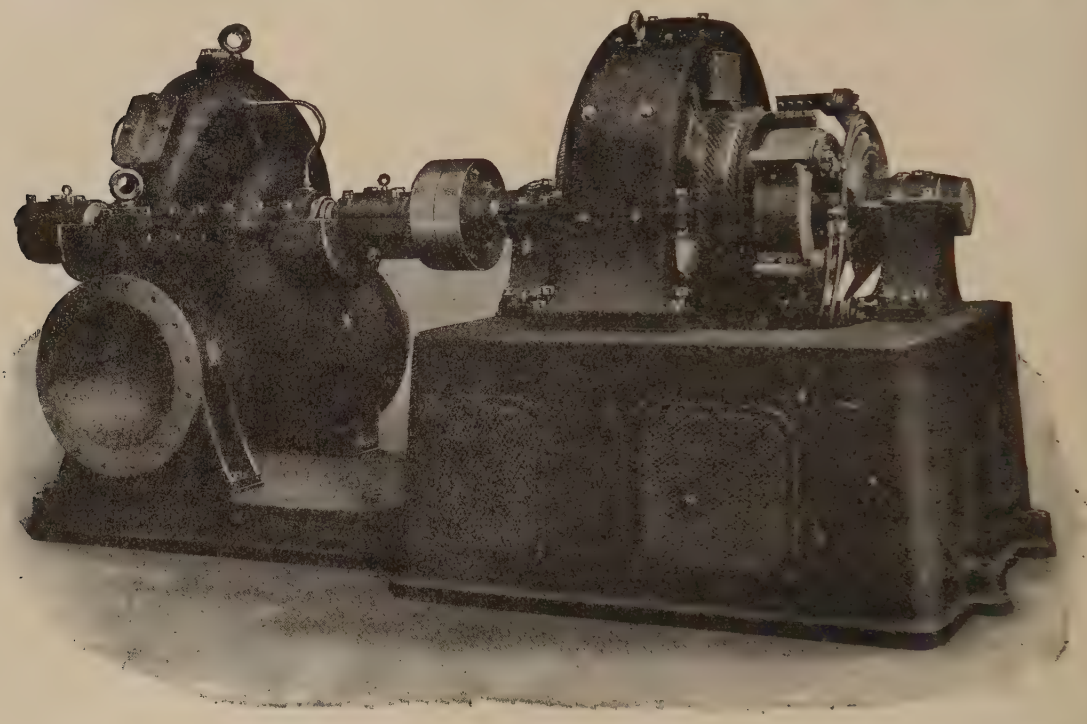


Fig. 108.—Low Lift Centrifugal Pump driven by a 240 h.p. Electric Motor at the Borough of Stepney Electricity Works.

packing that can be obtained. As a further precaution against leakage, a water seal is provided to each gland as shown in the drawing, a small pipe connecting the seal at the centre of the gland with the volute chamber. The form of these glands is clearly shown in Fig. 106, which is a view of the complete rotor of the pump. The form of the impeller is also very clearly shown in this illustration. Attention may also be drawn to the coupling bolts, which are provided with india-rubber bushes carried on bronze sleeves in order to provide a certain measure of flexibility between the pump shaft and that of the engine or motor which drives the pump. This coupling is so designed that a slight lateral movement is possible, so that when the pump is motor-driven, the armature can move to its correct magnetic centre. The bearings are of cast iron lined with white metal, split at the centre to facilitate removal of the shaft. The bearing surface is large in order to ensure cool running, and adequate lubrication is provided by means of light rings which dip into an oil bath arranged below the bearing. The bearings

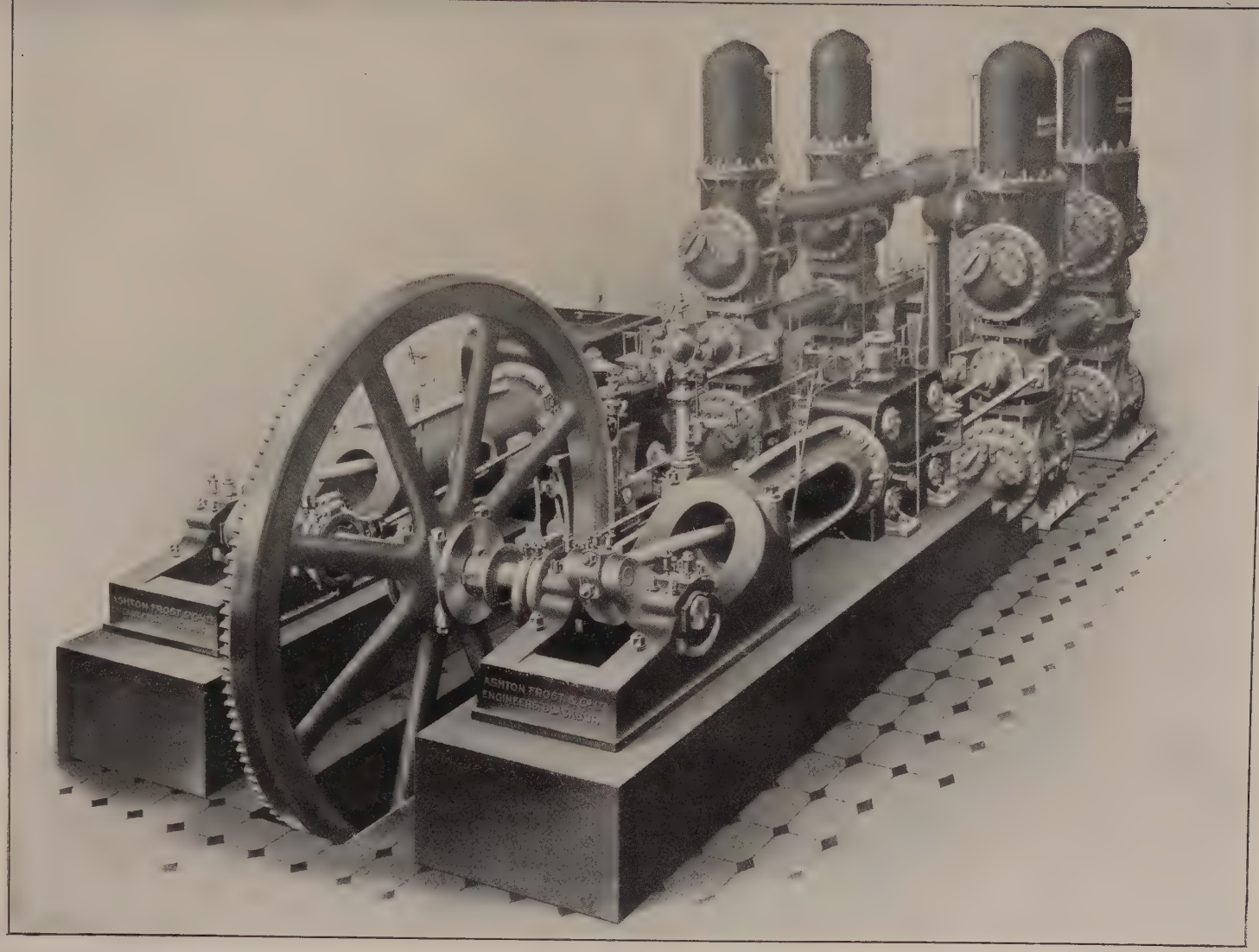


PLATE III.—HORIZONTAL COMPOUND ENGINE DRIVING DOUBLE-ACTING PUMPS DELIVERING 4,500,000 GALLONS OF WATER PER DAY AGAINST A HEAD OF 360 FT. DESIGNED AND CONSTRUCTED BY MESSRS. ASHTON, FROST & CO., LTD., FOR THE METROPOLITAN WATER BOARD.

themselves are carried on brackets of rigid design as shown in Fig. 104, although in the smaller units the bearings are bolted on to the side of the pump casing as shown in the sectional drawing, Fig. 105.

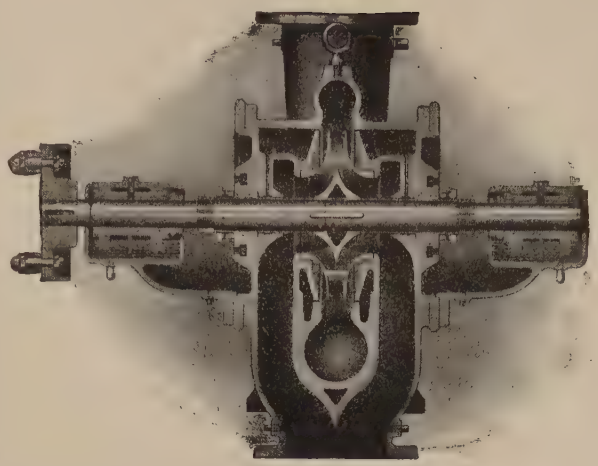


FIG. 109.—Sectional View of Medium Lift Centrifugal Pump by Messrs. Mather & Platt.

Some idea of the performance of these pumps will be gathered from the curves reproduced in Fig. 107, which are generally known as the characteristic curves of the pump, since they show the head at which the water is delivered, the b.h.p required to drive the pump, and its efficiency at different rates of discharge. The curves shown may be taken as being applicable to a pump having a normal discharge of 3,000 gallons per min. against a head of 55 ft. At this load the power required to drive the pump is 56 b.h.p. and the efficiency is 88 per cent. It will be noticed that, for both a greater and a lesser delivery, the efficiency falls, thus showing that the pump is operating under the most efficient conditions when delivering 3,000 gallons per min. At the maximum power of the driving motor, the speed of which is constant at 975 revs. per min., i.e. at 60 b.h.p., the pump delivers 3,800 gallons per min. against a head of 40 ft., the efficiency of the pump being 80 per cent. The efficiency curve shows that not only is the high efficiency maintained over a considerable range of delivery, but that the makers have succeeded in designing a pump of very high efficiency in view of the fact that only a simple volute discharge chamber is employed. In the majority of cases these pumps are direct coupled to electric motors, particularly when used in electric power stations. High-speed reciprocating engines are, however, sometimes employed, while occasionally for small deliveries a belt drive is employed. The general appearance of one of these pumps together with its driving motor will be gathered from Fig. 108, which shows a 24 in. pump supplied by Messrs. Mather & Platt, Ltd., to the Electricity Department of the Stepney Borough Council. The pump, which is designed to deliver 14,000 gallons of water per min. against a head of 42 ft., is driven by a 240 b.h.p. direct current motor having a speed of 560 revs. per min. The whole forms a very compact and efficient pumping unit.

Where medium lifts are required, that is to say, lifts somewhere in the region of 100 ft., particular care has to be paid to the design of the volute chamber, in order that

FIG. 110.—Shaft and Impeller of Mather & Platt Medium Lift Centrifugal Pump.

the kinetic energy of the water leaving the impeller may be converted into pressure energy equivalent to the desired head. Further, the speed of the impeller must be very high and its diameter large compared with the low lift type, so that the kinetic energy of the water when it leaves the impeller may be as high as possible. The principal features of the design of the Mather & Platt medium lift centrifugal pump will be gathered from a study of Fig. 109, which is a sectional view of a pump of this type. It will be noticed that the impeller is of the double-suction type drawing its water from both sides, thus eliminating the possibility of end thrust. The casing consists of a single casting, i.e. it is not split about the centre line as in the low lift pump; instead, two large end covers are provided which not only serve to support the bearing brackets, but which enable the impeller and shaft to be withdrawn bodily from the casing. A view of the shaft and the impeller of this pump is given in Fig. 110. In order to determine the best form of volute chamber for converting the velocity head into pressure head, Messrs. Mather & Platt recently carried out an exhaustive series of experiments, and the area of the volute at different sections is based upon the results of these experiments. The impeller is made of phosphor bronze with the vanes filed to template, it is mounted on a mild-steel shaft, the whole being carefully balanced to ensure smooth running. The glands where the shaft passes through the cover plates and the packing are of the same form as already described for the low lift pump, as also are the shaft bearings.

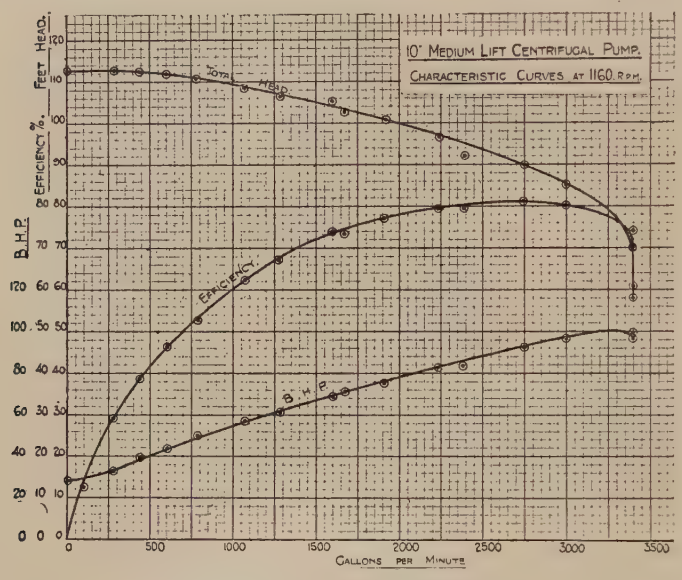


Fig. 111.—Characteristic Curves for a Mather & Platt Medium Lift Centrifugal Pump.

A good idea of the performance of this type of pump will be obtained from a study of the characteristic curves reproduced in Fig. 111. These were obtained during the trials of a 10 in. medium lift centrifugal pump driven at a speed of 1,160 revs. per min., and having a normal delivery of 2,750 gallons per min. against a head of 90 ft. It will be noticed from the curves that at this delivery the pump is working under conditions of maximum efficiency, although for a considerable range on either side of normal load, i.e. for low loads and for overloads, the efficiency is maintained at a high figure, as shown by the flatness of the efficiency curve. The maximum efficiency reaches 81 per cent., which is very good for the size of pump under consideration. The power delivered by the motor driving the pump at the above normal load is 92 b.h.p. It may be noted that it is impossible to overload the motor driving the pump above a certain predetermined amount, however much below the normal the head is reduced against which the pump works. This is revealed by the sudden collapse of the curves when the delivery exceeds 3,250 gallons per min. The standard overload usually provided is 10 to 15 per cent. above the power required to drive the pump at its normal rated output.

We come next to the consideration of what may be termed high lift pumps. It has already been explained that in order to obtain a high lift, the kinetic energy of the

water leaving the impeller must be converted into pressure energy, and that in order to do this with the maximum efficiency, an arrangement of guide vanes must be provided, these guide vanes forming a series of diverging passages and constituting what is termed the diffuser of the pump. Where the head required is not excessive, a single-stage pump can be employed, and Messrs. Mather & Platt have designed single-stage high lift turbine pumps to give heads up to 300 ft., although above this the multiple-stage pump is to be preferred. The main features of the design of this pump will be understood from a study of Fig. 112, which shows a sectional elevation taken through the centre of the impeller of a single-stage high lift pump. It will be noticed that the impeller is of the double-suction type, the water being drawn in at the centre on both sides and discharged at very high velocity at the periphery, the form of the impeller being very similar to that shown for the medium lift pump. On leaving the impeller, the water enters a diffuser consisting of a series of diverging passages formed by guide vanes carried on a ring which is mounted round the outside of the periphery of the impeller as shown. The form of the vanes inside this ring will be gathered from Fig. 113, which gives a good idea of the diverging passages between the guide vanes. This view really shows the guide vanes of a multi-stage pump in which the casing of the adjacent stage forms the other boundary of the guide vane ring, the ring in the single-stage high lift pump being bounded on both sides by discs which themselves are shaped so as to form a diverging passage as shown in transverse section (Fig. 112). The form of the guide vanes is, however, the same in both pumps, so that Fig. 113 may be considered as a section of the guide vane ring taken in the plane of rotation of the impeller.

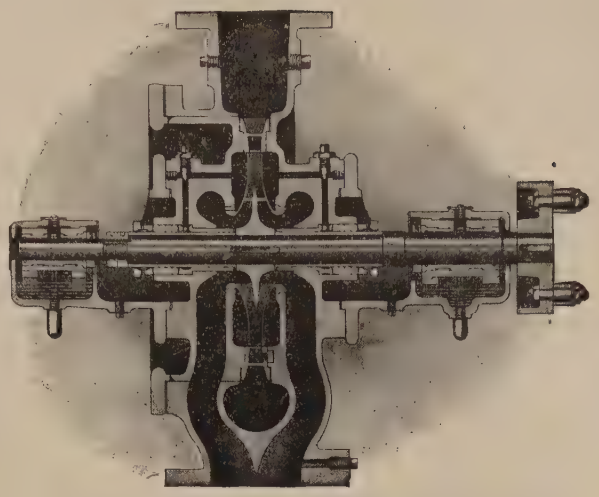


Fig. 112.—Sectional View of Single-stage High Lift Centrifugal Pump by Messrs. Mather & Platt.

Fig. 113.—Guide Vanes or Diffuser for High Lift Centrifugal Pump.

It is to be noted that the angle of the guide vanes is such that the water enters the diffuser without shock, and that the areas of the passages have been accurately determined as the result of experiments so that the kinetic energy of the water entering the ring is converted into pressure energy with the minimum of loss. To assist in this direction, the guide vanes are carefully finished to a smooth surface in order to reduce the loss due to friction. On leaving the diffuser, the water, now at considerable pressure and therefore at comparatively low velocity, enters the volute chamber, which again is of diverging form, so that more of its kinetic energy is converted

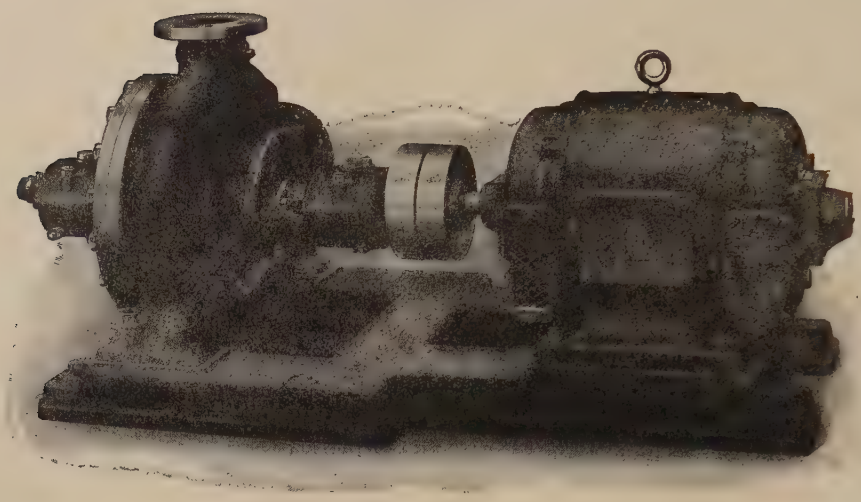


Fig. 114.—Single-stage High Lift Centrifugal Pump direct coupled to an Alternating Current Electric Motor.

into pressure energy, and thus when it reaches the discharge flange at the top of the pump the water is at a comparatively high pressure, i.e. it possesses a high lift. The use of the diffuser possesses the double advantage that not only does it convert the kinetic energy of the water into pressure energy, but it enables the water to enter the volute chamber at comparatively low velocity, and hence the losses in this chamber due to friction and eddy currents are reduced to a minimum, or, in other words, the efficiency of the pump is very high. The principal features of the design are very similar to those for the medium lift pump, but attention may be drawn to the arrangement of the end covers. It will be noticed on the right of Fig. 112 that a small cover is provided which serves to carry the inner bearing, i.e. the bearing on the shaft coupling side, while on the other side a larger cover is fitted, which not only provides a space in the casing through which the impeller can be withdrawn, but which keeps in position a casting which serves to keep the guide vane ring in position. Further, it gives the correct form to the left-hand suction branch of the pump, while it also serves to take the water seal pipes to the glands. This arrangement is called for, since provision has to be made for inserting and removing (when necessary) the guide vane ring. The relative size and position of these covers will be readily understood from Fig. 114, which shows a single-stage

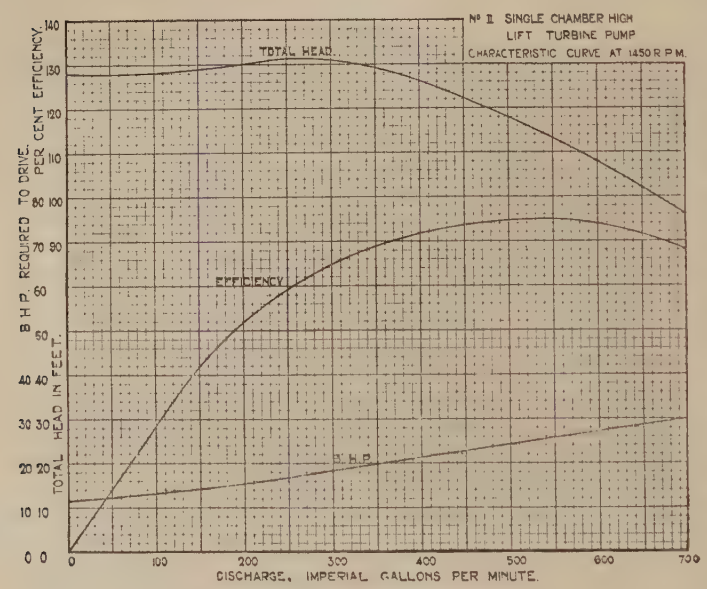


Fig. 115.—Characteristic Curves for a Mather & Platt Single-stage High Lift Centrifugal Pump.

high lift pump direct coupled to an alternating current motor. This view also gives a good idea of the substantial size of the bearings, as well as of the general appearance of sturdiness of the complete pump.

The characteristic curves of one of the Mather & Platt single-stage high lift pumps are given in Fig. 115. These were taken for a pump having a normal running speed of 1,450 revs. per min. It will be noticed that when running at its maximum efficiency, the pump delivers about 550 gallons of water per min. against a total head of 114 ft., the power required to drive the pump being then 26 b.h.p. The maximum head against which the pump will work is 132 ft., which, as the student will be able to figure out for himself, is equivalent to a pressure at the pump discharge flange of about 57 lb. per sq. in. At this head the efficiency is somewhat lower, as might be expected. Even so, it reaches 63 per cent., which is a very good performance for a high lift centrifugal pump.

When still higher pressures than that represented by a head of 300 ft. (say 130 lb. per sq. in.) are required, then the single-stage pump is no longer suitable and we have to employ a number of stages arranged in series on the same shaft. The principle under

FIG. 116.—Four-stage High Lift Centrifugal Pump direct coupled to an Electric Motor.

which the pump works is, however, precisely the same as that given above. The impeller of the first stage gives a considerable increase in the kinetic energy of the water, which is then converted into pressure energy in the first diffuser. From the first diffuser the water is delivered to the impeller of the second stage, from which it is discharged at high velocity into the second diffuser, where its pressure is further increased. This occurs in each successive stage, so that the pressure of the water is progressively raised as it passes through the pump. The general appearance of a multi-stage high lift turbine pump will be gathered from Fig. 116, while the principal features of the design will be understood from Fig. 117, which shows a sectional elevation of the same pump turned the other way round. It will be noticed that the pump is built up of four stages, each stage consisting of an impeller keyed to the shaft, a diffuser or guide vane ring mounted around the periphery of the impeller, and a casing of special volute form through which the water passes from the periphery of the guide vane ring down to the centre of the next impeller. The casings are held together by two heavy end plates, one of which carries the inlet pipe and the other the outlet pipe, and the whole is rigidly secured together by a number of longitudinal bolts running the entire length of the pump body. It is to be noted that the casings of each stage are so formed that when bolted together they give just the requisite clearance for the impellers which rotate between them.

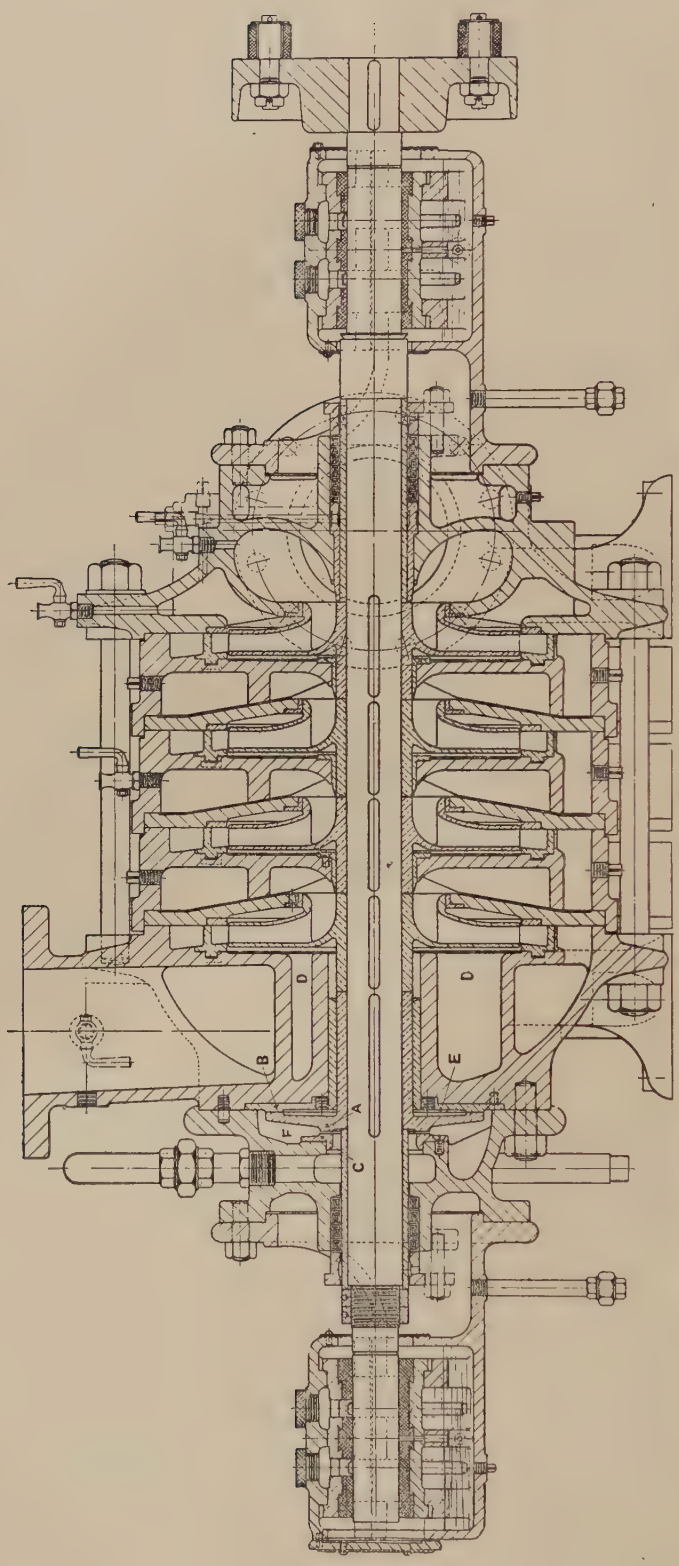


Fig. 117.—Sectional Elevation of Four-stage High Lift Centrifugal Pump by Messrs. Mather & Platt, Ltd.

It will be observed, from a study of the drawing that the impellers are of the single suction type, and thus, while the amount of leakage in the clearance spaces on either side of the impellers is about the same, the area subjected to the pressure of the leakage water is greater on the delivery side of the impeller than it is on the suction side. Hence there is a thrust towards the suction end of the pump the magnitude of which is equal to the difference of these areas multiplied by the pressure in the clearance spaces and multiplied by the number of impellers. With large units delivering water at great pressure this end thrust may exceed 10 tons, and at the high speeds at which these pumps are run it is a very difficult matter to obtain a satisfactory design of thrust block for taking up this end thrust. After much experimenting, Messrs. Mather & Platt evolved an automatic disc balancing device which has proved entirely satisfactory, and which in one form or another is now to be found on almost all high lift multi-stage turbine pumps. The principle of this balance valve device will be easily understood from a study of the drawing. The balance valve is shown at A resting on a

large renewable seating B. The diameter of the valve is calculated so that its area when acted upon by the pressure of the water in E (which is equal to the pressure at the last stage of the pump less the loss of pressure due to the flow of the balance water between the hub of the valve A and the bushing) is much greater than that required to overcome the end thrust due to the impellers. Assume that when the pump is stationary, the balance valve face is bearing on the large seating B. At starting up, as soon as pressure is developed in the pump, this pressure will be communicated to the space E and the balance valve will move towards the small seating C. The result is the leakage of water from the space E into the space F, which is connected to the suction side of the pump and is therefore at low pressure, thus reducing the pressure in the space E. As the valve continues to move over towards the small seating C, the amount of leakage increases until the position is reached when the pressure in E is just sufficient to balance the end thrust due to the impellers. The balance valve then floats between the large and small seatings and maintains perfect equilibrium, there being a steady flow of water

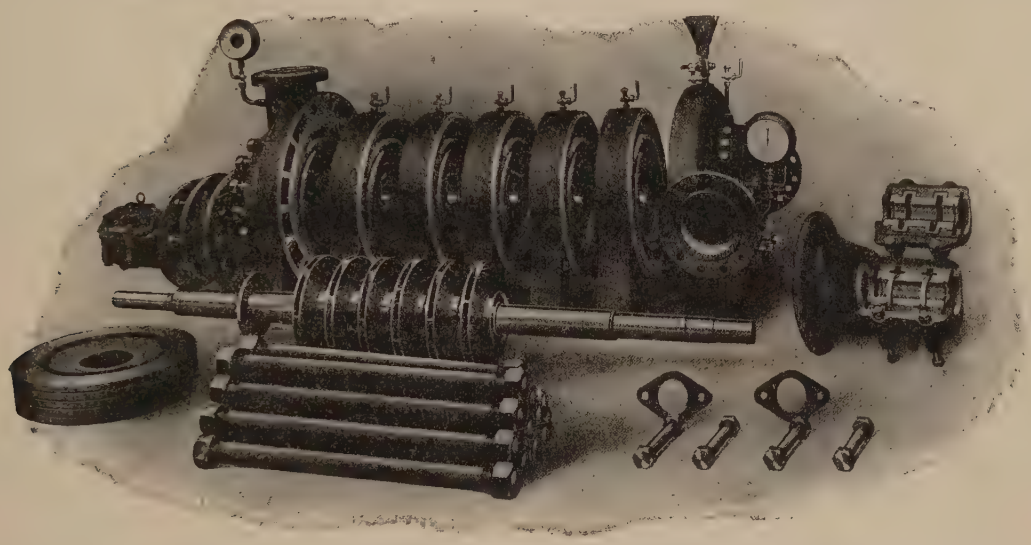


Fig. 118.—The Various Parts of a Mather & Platt Six-stage High Lift Centrifugal Pump previous to assembly.

through it all the time the pump is working. When clean water is used this involves little or no wear on the valve faces, etc., but even with gritty water the amount of wear is not excessive, and the various parts can be renewed both easily and cheaply.

The various other parts of the pump call for little or no attention. The end covers, which carry the suction and delivery connections, are fitted with glands where the shaft passes through so as to prevent leakage of water from the body of the pump. The gland at the suction end is water-sealed, but owing to the balance valve at the delivery end no seal is required in the gland at this end, the water which passes into the space F finding its way to the suction side of the pump as already indicated. These end covers also serve to carry the brackets in which the bearings are housed, these latter being of cast iron white metal lined, and fitted with ring oil lubricators. The form of these bearings, as well as of the other parts of the pump, will be gathered from Fig. 118, which shows the various parts of a six-stage high lift pump prior to assembly. Particular attention may be drawn to the six impellers mounted on the shaft and to the guide vanes, which can be clearly seen in the various segments of the pump casing. Attention may also be drawn to the funnel provided on the suction end cover. This is for filling the pump

with water before starting up, a precaution which has to be taken with all forms of centrifugal pumps. Wherever possible, the pump is arranged below the suction head so that the pump is naturally full of water even when it is not in action. If, however, it is not possible to arrange for this, the pump must be "primed," i.e. filled with water, from some outside source as indicated. The nature of the performance of one of these multi-stage high lift pumps will be understood from a study of the characteristic curves reproduced in Fig. 119, which were obtained from the tests of a pump designed to deliver 600 gallons of water per min. against a head of 700 ft., the power required to drive the pump being 165 b.h.p. The speed of the pump is 1,460 revs. per min., and the efficiency under these conditions reaches the high value of 76 per cent. Moreover, the efficiency is maintained at over 70 per cent. for a range of delivery from 400 gallons to 850 gallons per min.

In order to obtain the highest pressures such as are employed in operating hydraulic machinery instead of having a single multi-stage pump, two of these units can be arranged in series, the delivery from the first unit becoming the suction for the second. Thus if, as is sometimes the case, we have ten stages connected together to form a single unit and two of these units are arranged in series, we have the equivalent of a twenty-stage pump. The pumps are mounted on the same bedplate with the driving motor between them, the whole forming a remarkably compact unit. Some idea of the appearance of these units will be gathered from the views of the pumps reproduced in Plate IV, both of which have been designed and constructed by Messrs. Mather & Platt, Ltd. The pump shown in the upper view is, it is claimed, the largest single-unit hydraulic pumping plant in the world. It is designed to deliver 800 gallons of water per min. at a pressure of 1,500 lb. per sq. in., the motor, which is of 1,400 h.p., having a speed of 1,450 revs. per min. This plant has been installed in a large works in the North of England, and is used to supply power for operating various hydraulic machinery. The unit shown below also has a capacity of 800 gallons per min., but here the working pressure is not so high, being only one-half of that of the unit shown above, viz. 750 lb. per sq. in. In the upper unit the various stages are enclosed in cast-iron casings, but the lower unit follows the practice already described, and it will be observed that each pump consists of six stages. The driving motor is of 650 h.p. and has a speed of 1,480 revs. per min. Attention may also be drawn to the installation shown in Figs. 120 and 121, which are reproductions from photographs of two hydraulic pressure pumps installed by Messrs. Mather & Platt at Prince's Dock, Glasgow, for the Clyde Navigation Trust. These pumps each have a capacity of 400 gallons per min. at a pressure of 750 lb. per sq. in., and are driven by electric motors each of 350 b.h.p. and supplied with current at 6,500 volts. It is of interest to note that these rotary pumps have displaced the steam reciprocating sets.

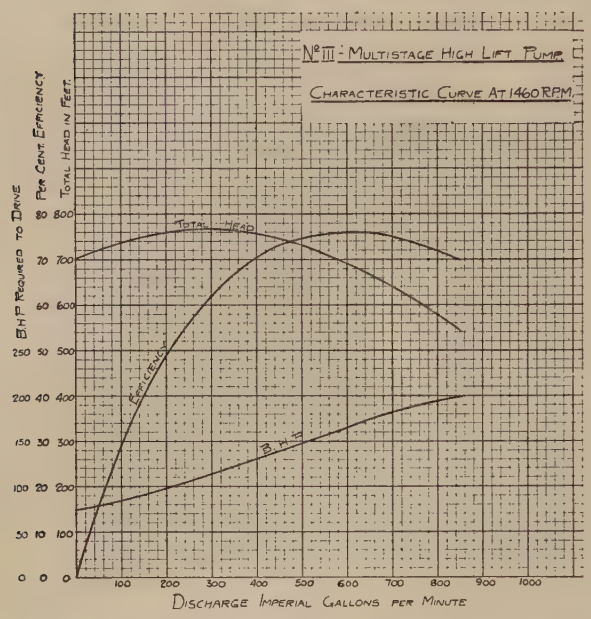


Fig. 119.—Characteristic Curve for Mather & Platt Multi-stage High Lift Centrifugal Pump.

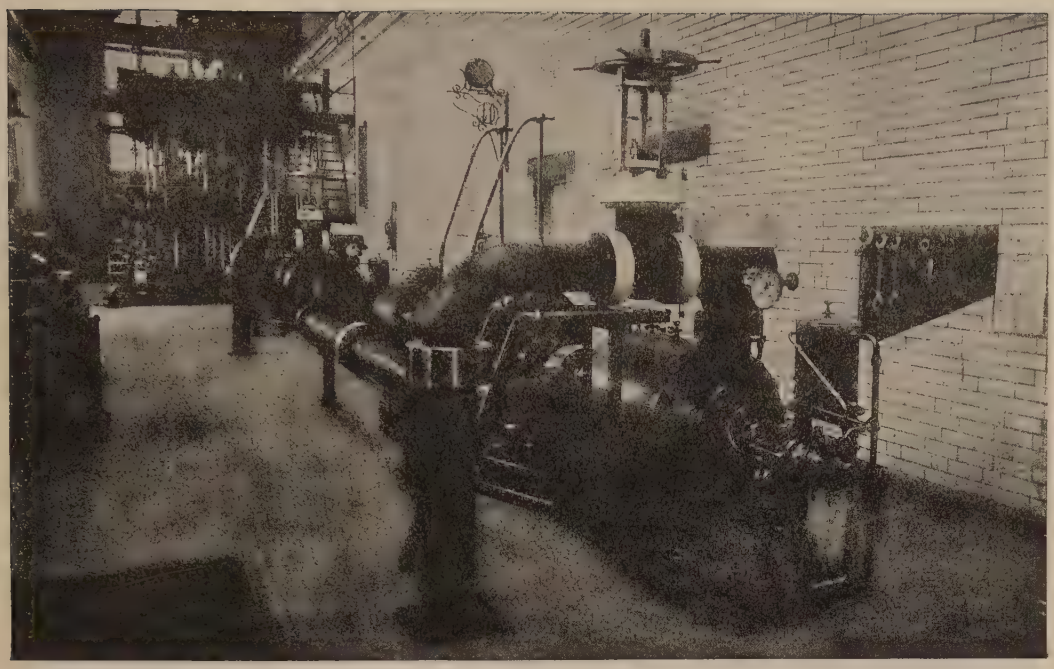


FIG. 120.—Two Rotary Hydraulic Pumps by Messrs. Mather & Platt, Ltd. Installed at Prince's Dock, Glasgow.

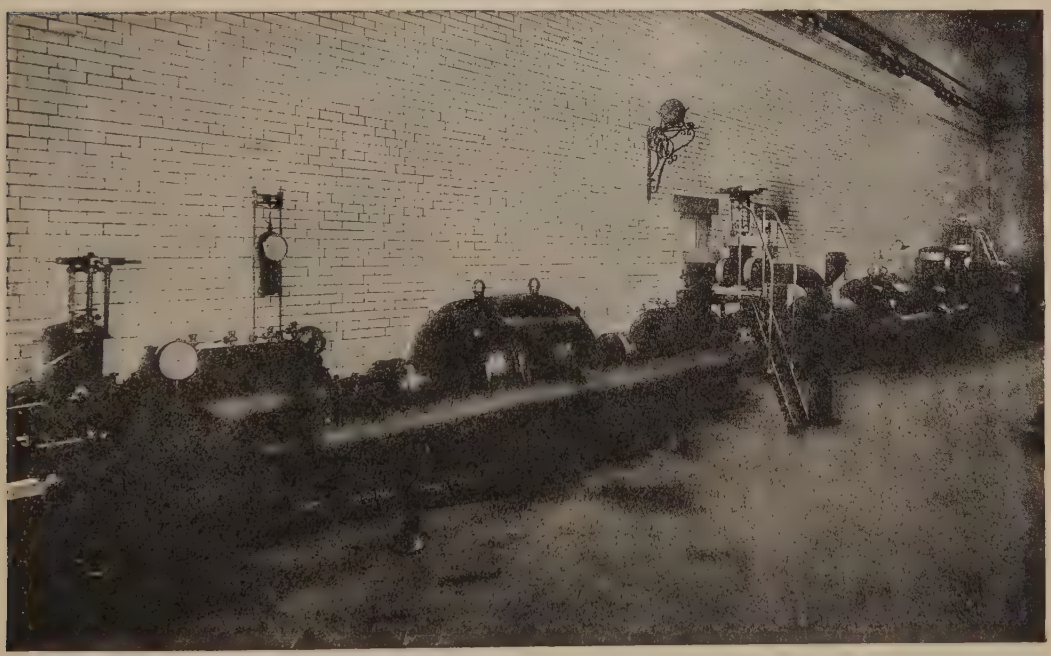


FIG. 121.—Another View of the Two Pumps shown above. Each Pump has a capacity of 400 gallons per min. at a pressure of 750 lb. per sq. in.

seen in the background of Fig. 120, and the latter will be ultimately removed. Another interesting installation is that shown in Fig. 122, which is a reproduction of a photograph of the pumping station of the London Hydraulic Co., Ltd. In the background will be seen three triple-expansion steam engines driving hydraulic pumps of the reciprocating type, while in the foreground is a tandem rotary pump driven by an electric motor

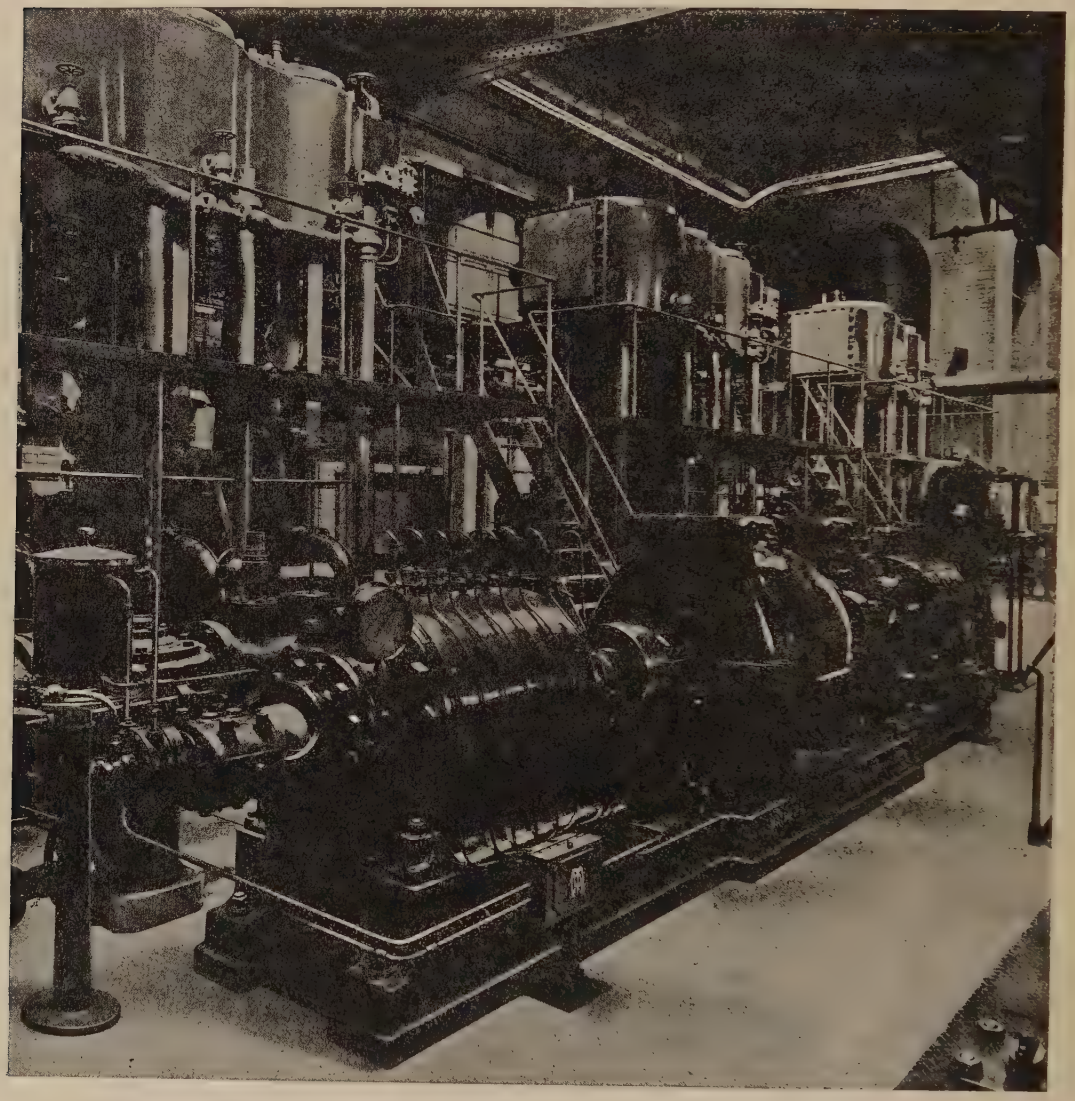


FIG. 122.—Tandem Turbine Pumping Unit installed at the Pumping Station of the London Hydraulic Co., Ltd. Capacity, 500 gallons per min. at a pressure of 800 lb. per sq. in.

mounted between the two portions of the pump, these being connected together in series. The pump, which was supplied by Messrs. Mather & Platt, Ltd., has been designed for a duty of 500 gallons per min. at a pressure of 800 lb. per sq. in., the motor developing 440 b.h.p. at a speed of 1,480 revs. per min. At the official test the actual pump efficiency realised was 75 per cent., which must be regarded as a very satisfactory performance. An interesting comparison can be made from these illustrations of the reciprocating and the rotary types of hydraulic pump.

The above important examples of the use of centrifugal pumps are sufficient to indicate that this type is regarded as being capable of performing any kind of pumping work, whether against very high heads producing water under considerable pressure for driving hydraulic machinery, or against low heads discharging enormous volumes of water from docks, etc. It will be recognised from the figures which have been given that the efficiency of the centrifugal pump is not so high as that of the reciprocating pump, but, on the other hand, the cost of the former, its weight and the space it occupies for a given output, are considerably less than that of the latter. Moreover, the cost of maintenance and upkeep is also less. In the reciprocating pump—including the engine which drives it—there is a large number of working parts involving a correspondingly large number of bearing surfaces at all of which wear is taking place and which sooner or later require renewal. There are in addition a number of valves in the pump itself, as well as packings in pump plungers and pump casing stuffing boxes, all of which gradually tend to leak and which must be kept in a thoroughly water-tight condition if the high efficiency of the pump is to be maintained. On the other hand, the centrifugal pump consists of only one moving part with no valves and only two glands where the impeller shaft passes through the casing. It is true that the pump is rotating at very high speed, and therefore the wear and tear on the packing on the glands, as well as on the two bearings, is considerable, but if these are efficiently designed and well lubricated, the cost of maintenance and renewal should not be very great.

There is one other direction in which the centrifugal pump is superior to the reciprocator. The student will possibly have remarked in studying the photographs of installations of rotary pumps that no air vessels are shown, although these form a very prominent feature of the installations of reciprocating pumps. The reason is that such air vessels are not called for, as the delivery from the rotary pump is perfectly continuous. This means that the vibration and the possibilities of fracture in the various parts of the installation due to water hammer are almost entirely eliminated. If a centrifugal pump is running at full speed and water is being delivered at normal head, then if the delivery valve is closed, it will be found that beyond a slight increase in the pressure inside the pump casing nothing will occur, the impeller merely churning up the water in the casing. Further, it will be found that the load on the motor driving the pump is considerably reduced, actual tests showing that only about one-third of the current is required with the delivery valve shut compared with what is required when the pump is working at full load. It is because of the capability of the rotary pump to adjust itself to wide variations of the load that in hydraulic power installations no accumulator is required unless the load varies over a very wide range. Even then it is not so much because the turbine pump could not cope with a wide fluctuation of load as that it is then unnecessary to install pumps capable of dealing with the maximum load. If an accumulator is installed, the capacity of the plant is equivalent to the mean load, but if the fluctuation is not more than, say, 30 per cent. on either side of the normal, it is not considered advisable to fit an accumulator. On the other hand, the centrifugal pump, since it is not positive in its action, suffers from the disadvantage that it may "lose its water." We have drawn attention to the fact that the pump must, if possible, be placed below the suction inlet, or that some provision must be made for flooding the pump before starting up. While, theoretically speaking, the pump should be capable of drawing water through a suction head of about 34 ft., actually the impeller churns up the air and the suction lift is very small. This means that if the casing is empty, it is a difficult matter to get the pump to commence delivering water. A small " snifting " valve is generally fitted on the casing which is opened on starting up the pump, and through this any air in the casing can be discharged, the valve also serving to show when the pump is delivering water. While, however, the loss of water may sometimes be a source of trouble when dealing with

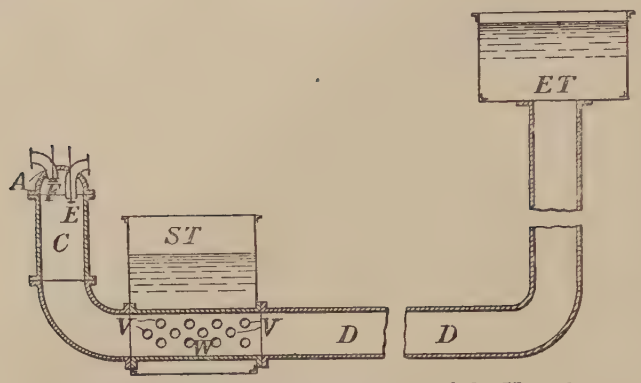


FIG. 123.—Diagrammatic Arrangement of the Humphrey Gas Pump.

centrifugal pumps, if the pump is installed in such a position that it is always flooded, or if the precaution is taken of filling the casing before starting, little trouble should ever be experienced.

If the rotary pump is remarkable for its simplicity, what must be thought of a pump in which there are no moving parts! We have already drawn attention to one such pump during the course of our brief historical survey of the development of the steam engine,[1] viz. the pulsometer steam pump. The pulsometer steam pump, apart altogether from its practical use, is of interest because of its resemblance to the engine devised by Savery. It will be recalled that this pump has neither pistons nor impellers, the movement of the water from the suction to the delivery being caused by the pressure of steam on the free surface of the water in the body of the pump, the actual flow of water being regulated by valves. It is apparent that, instead of steam pressure, we might employ almost any other fluid pressure, including the pressure due to the explosion of a mixture of air and gas. This is what is done in the "Humphrey Gas Pump," a type of pump which is being successfully employed on a commercial scale for raising large volumes of water through a comparatively low head. The Humphrey pump is now being developed on a commercial scale by the world-famous firm of engineers and shipbuilders, Messrs. Beardmore & Co., Ltd., and we are indebted to them for the following particulars of the installation at the Chingford Reservoir of the Metropolitan Water Board, which is, at the time of writing, the largest installation of Humphrey pumps that has so far been laid down. Some idea of the magnitude of the installation will be gathered from the fact that the pumps, of which there are five, are designed to pump 180,000,000 gallons of water per day of 24 hours, from the River Lea into the King George Reservoir, the lift being 25 to 30 ft.

Before describing in greater detail the pumps installed at Chingford, we may pause for a moment to explain briefly the principle upon which the pump works. A diagrammatic view of the pump is shown in Fig. 123, from which it will be seen that it consists essentially of an internal combustion cylinder C, connected by means of a long pipe D, of the same diameter as the cylinder, to a delivery tank ET, a suction tank ST being arranged near the internal combustion cylinder. Imagine now that the system contains water and that a mixture of air and gas is exploded in the cylinder C. The pressure thus developed acts upon the surface of the water and sets the whole column of water in D in motion, the motion continuing as the burnt gases in C expand. Once, however, the water is set in motion, it continues to move owing to its own momentum until the pressure in the combustion cylinder falls below that of the atmo-

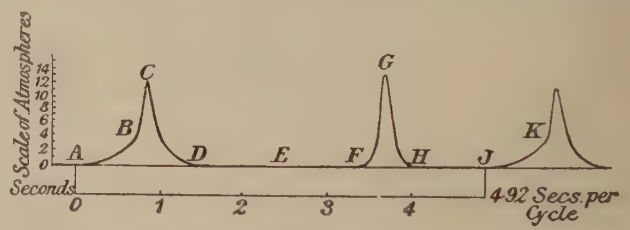


FIG. 124.—Indicator Diagram showing Fluctuation of Pressure in Combustion Cylinder of the Humphrey Gas Pump.

[1] See Volume I, Chapter III.

sphere, when the exhaust valve E opens under its own weight. Moreover, as the water moves forward under the impulse of the explosion, the time soon comes when the level of the water in C falls below that in the suction tank ST and more water is then able to enter the system through the valves V. At the same time a scavenging valve in the cylinder head—not shown in Fig. 123—opens and pure air is drawn into the combustion space. The momentum of the water is ultimately exhausted, and like a pendulum it begins to move backward in the pipe D. This motion sweeps the burnt gases through the exhaust valve E, which remains open until the level of the water reaches the valve, when the impact causes the valve to shut, thus entrapping a mixture

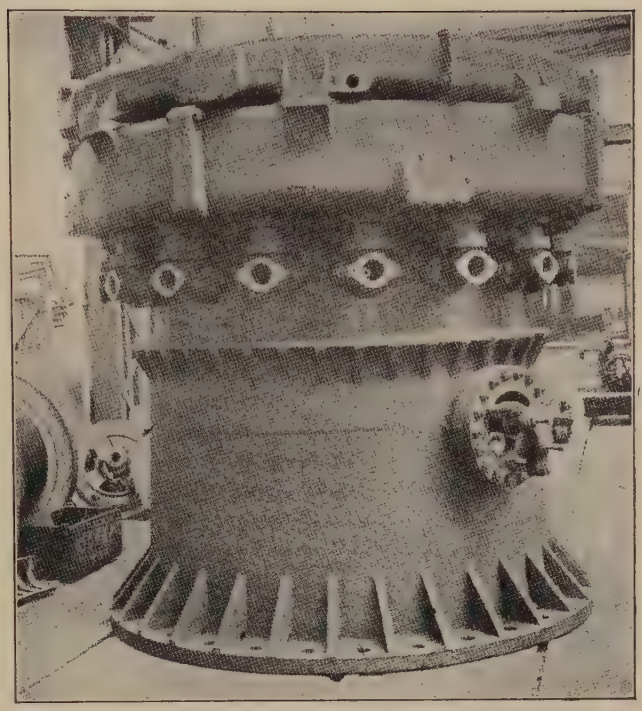


FIG. 125.—Combustion Cylinder, 7 ft. internal diameter, for Humphrey Gas Pump.

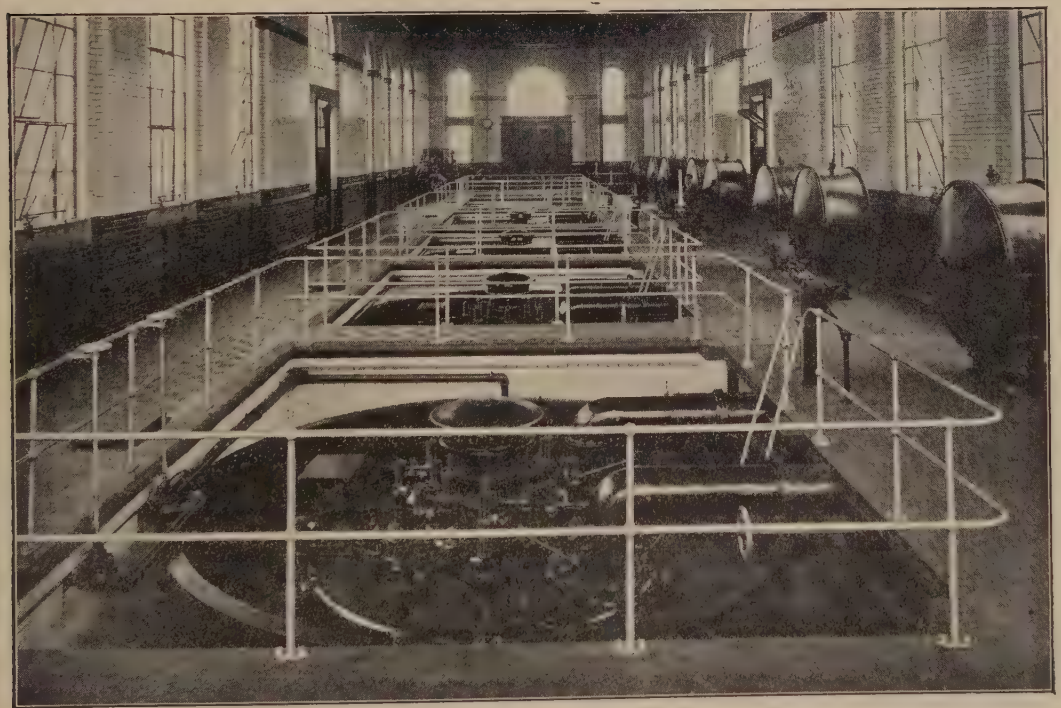


FIG. 126.—Interior of the Engine Room of the Chingford Pumping Station of the Metropolitan Water Board, showing the Cylinder Heads of the Five Humphrey Gas Pumps.

of exhaust gas and scavenging air in the space F, which is cushioned by the upward movement of the water in C. Ultimately, the water again moves forward and the pressure soon falls below atmospheric. By means of a locking device, the exhaust valve E and the scavenging valve are prevented from opening, while the valve A is unlocked so that a charge of air and gas is drawn into the combustion chamber. When the forward momentum of the water is again exhausted the water moves back, closing the valve A and compressing the mixture of air and gas in C. At the end of the backward flow, i.e. when the mixture is fully compressed, it is automatically exploded and the cycle of operations commences again.

It is to be noted that the valves V open inwards, so that, while as a result of the backward and forward flow of the water through W water is drawn into the system, it cannot

FIG. 127.—The King George Reservoir at Chingford, showing the Discharge from the Humphrey Gas Pumps.

flow back again into the suction tank, but is discharged to the delivery tank, whence it overflows with each forward movement of the water column and is thus discharged from the system. The cycle of operations in the combustion cylinder is obviously the same as that in the cylinder of a four-stroke internal combustion engine, although by the provision of scavenging pumps it would be possible to operate the pump on the two-stroke cycle. Indicator diagrams can be easily taken during the cycle of operations if the indicator drum is rotated by clockwork, the result obtained being similar to that shown in Fig. 124. In this diagram AB represents the compression of the mixture of air and gas. At B the charge is fired, the pressure rise being indicated by BC. The burnt gases then expand until the pressure falls to atmospheric at D, when the exhaust valve opens. The pressure in the cylinder remains practically constant until at F the exhaust valve closes and the cushioning occurs as shown by FG. The water is driven for-

ward as a result of this cushioning, drawing in the charge of air and gas from H to J, until with the return movement of the water compression occurs and the cycle is repeated. Actually it is found that the cushioning pressure exceeds the explosion pressure, thus ensuring that a good charge of air and gas is drawn into the cylinder on the suction stroke. In the Chingford pumps, the maximum explosion pressure is about 12 atmospheres (say 176 lb. per sq. in.), while the cushioning pressure reaches 13½ atmospheres. It may also be noted that the time of the complete cycle is 4·92 secs., which is equivalent to just over 12 complete cycles per sec.

As already noted, there are five of these pumps in the installation at Chingford, four of them having cylinders 7 ft. diameter and capable of raising 40,000,000 gallons of water per day, while the fifth, with a cylinder 5 ft. diameter, is rated at 20,000,000 gallons per day. The general form of one of these cylinders will be understood from Fig. 125, which shows the cylinder casting for one of the 40,000,000 gallon pumps in the makers' shops. It may be noticed that, instead of having only one admission and one exhaust valve, a large number of smaller valves are fitted, the admission valves being arranged in the cylinder head and the exhaust valves at the facings shown around the top of the cylinder. These valves are shown in Fig. 126, which is a reproduction of a photograph of the interior of the engine room at Chingford. The cylinder heads of the five pumps are just visible, that in the foreground showing clearly the admission valves. The pipes shown on the right are for supplying the gas to the cylinders, the flat cylinders against the wall being the gas bags which ensure a uniform supply during each suction stroke. The water pipes leading from the combustion chamber are each 6 ft. diameter and are built up of steel plates. These terminate in large vertical delivery chambers which serve as air vessels and to which the discharge pipes are secured, the head over the discharge outlet ensuring a practically uniform discharge. A good idea of the actual discharge from these pumps is obtained from Fig. 127, which shows the outlet pipes from the pumps at the side of the King George Reservoir at Chingford. The gas for operating the pumps is obtained from a battery of four Dowson pressure type gas producers using anthracite as fuel. A description of producers of this type, together with the results of tests of this actual installation, has already been given in that section of our work dealing with gas producers,[1] so that any further description is unnecessary.

The following table gives a summary of the results of a number of tests which were carried out to determine the output and the efficiency of the pumps at Chingford. Nos. 1 to 4 are the larger pumps rated at 40,000,000 gallons per day (27,800 gallons per min.), while No. 5 is the 20,000,000 gallon pump (13,900 gallons per min.).

SUMMARY OF RESULTS OF TESTS OF HUMPHREY GAS PUMPS AT THE CHINGFORD PUMPING STATION OF THE METROPOLITAN WATER BOARD

Pump number :	1	2	3	4	5
Number of tests	6	6	6	6	6
Average duration of tests . . . mins.	9·27	8·95	8·37	9·67	10·0
Lift . . . . . ft.	30·01	30·24	30·06	32·6	30·24
Water pumped . . . gallons per min.	33,497	32,773	33,047	32,663	18,116
Pump horse power . . . p.h.p.	303·9	300·4	301·1	322·7	166·0
Gas consumed per minute (measured at 60° F. and 30 in. mercury) . . cu. ft.	395·4	393·3	391·5	400·1	191·6
Calorific value of gas . B.Th.U. per cu. ft.	145·7	146·4	146·2	142·2	138·1
Overall efficiency . . . per cent.	22·39	22·19	22·33	24·07	26·63
Anthracite burned per p.h.p. per hour lb.	·946	·957	·949	·881	·796

[1] See Volume II, Chapter V.

The figures for overall efficiency give the ratio of the energy utilised in raising the water to the energy supplied to the pumps with the gas in the form of heat, and they prove conclusively that from the point of view of fuel consumption the Humphrey pump is very economical. At the same time the original cost of the installation at Chingford was found to be lower than that of an installation of either reciprocating or centrifugal pumps of the same output, this, of course, being the principal reason which led the Metropolitan Water Board to adopt this plant. It remains to be seen whether, as a result of continual operation, they will prove as economical from the point of view of maintenance and overhaul as they undoubtedly are from the point of view of original cost and fuel consumption.

CHAPTER V

THE MARINE ENGINE—TRIPLE- AND QUADRUPLE-EXPANSION RECIPROCATING ENGINES

MUCH has been written on the subject of the supreme importance of the British mercantile marine to the people of these islands. Certain it is, that without the ships of the mercantile marine this country would very rapidly approach the verge of starvation. With our large population the overwhelming proportion of which is engaged upon industrial and manufacturing pursuits, the food produced by that comparatively small proportion which is engaged in agriculture is far from being sufficient to meet our requirements. Hence an enormous amount of foodstuffs, notably grain and meat, has to be imported into these islands. Moreover, our industries are largely dependent upon supplies from overseas for the raw materials which they convert into finished goods, and thus we find that cotton, wool, and various ores are imported into this country in large volumes. The products of our industries are, however, far more than sufficient to meet the requirements of our people at home, and hence vast quantities of manufactured goods are exported to the dominions and to foreign countries all over the world. In addition, this country is fortunate in having an enormous wealth of coal beneath its soil, and not a small proportion of our export trade is represented by the coal which is shipped abroad. It may be mentioned that as far as possible a shipowner always endeavours to obtain a cargo for his vessels both on their outward and their home voyage, a journey in the light condition or in ballast being obviously uneconomical, and thus there is always a tendency for imports and exports to balance.

We are not concerned here with questions of politics, but it will be readily understood that this dependence upon outside sources and the necessity for finding markets for our manufactured goods has had a profound influence upon our national development. What we are more concerned with here, is the undoubted fact that the demand for ships in which this trade can be effectively carried on has had far-reaching effects upon engineering progress in this country. It is impossible to say whether general engineering practice has had more influence on marine engineering practice than has marine engine practice on general engineering practice, but it is certain that they have had considerable mutual influence on each other. Sometimes a development in land practice has been followed by a similar development in marine practice, and at other times progress in one branch of marine engineering has led to corresponding progress in the same branch of general engineering. It must be remembered in this connection that the steam engine was first applied to the driving of pumps on land, and at first the progress which was made was all in the direction of rendering it more suitable for various purposes on land. Later on, say at about the middle of last century, as the value of the steam engine for driving ships became to be generally recognised, progress in marine engineering was very rapid, until at the close of last century the finest products of the various marine engineering shops were equal, if not superior, to anything which was produced for use on land.

The advent of the steam turbine tended again to restore things to what they were before. From the marine point of view, the early steam turbine suffered from two great disadvantages—first, that it ran at very high speed, and second that it could not be reversed, both of which were not of any serious consequence in land practice, particularly the latter, since for the majority of purposes, notably for driving electrical generators, it is not necessary that the turbine should be able to run in either direction. The former disadvantage limited the application of the steam turbine to those vessels in which high speeds were required and in which the propeller could be driven at a tolerably high speed of rotation without too great a loss of efficiency. It may be noted that the economical speed of the propeller of a ship varies with the speed of the ship itself, but in no case can the former be very high. Thus with the majority of cargo vessels in which a speed of 10 to 12 knots[1] is considered most economical, the speed of the propeller is about 70 to 75 revs. per min.; on a high-speed liner like the *Mauretania*, with a speed of 25½ knots, the propellers turn at about 175 revs. per min., while for a fast torpedo boat destroyer, in which the speed may reach anything from 36 to 40 knots, the speed of the propellers is 240 revs. per min. Consider then what the conditions must have been in early turbine propelled vessels in which the turbines were direct coupled to the shaft. The writer had the opportunity of taking part in the trial trips of one of the first experimental torpedo boat destroyers which were fitted with steam turbines. The turbine had a speed of 1,250 revs. per min.; and despite most careful designing of the propellers, the "slip" amounted to 54 per cent., or in other words, one-half of the energy of the turbine was employed by the propeller in merely stirring up the water. It gradually became possible, by using turbines of large size, to bring down the speed of the turbine considerably, although the turbines of the *Mauretania*, revolving at 175 revs. per min., must be considered as exceptional on account of their slow speed. Generally speaking, the speed was brought down to somewhere in the region of about 500 revs. per min., and thus, except for high-speed vessels, the turbine could not be considered an economical form of prime mover, at least from the point of view of efficiency of the propeller.

Apart, however, from the question of propeller speed, the question of the size and weight of the turbine is also one of considerable importance, particularly on board ship, where the space occupied by the machinery must be kept as small as possible in order that the space available for the carriage of cargo shall be as large as possible, and where the weight of the machinery is so much dead weight which always has to be carried about with the ship. We have already seen, in that section of our work dealing with steam turbines,[2] that one of the chief characteristics of this form of prime mover is its high speed. We might almost lay it down as an axiom that the higher the speed the more efficient the turbine. Thus the reduction of the speed to suit the special requirements of the propeller must always be accompanied by a loss of efficiency. Moreover, for a given power, the employment of a low speed of rotation necessitates the use of a turbine of greater diameter and of greater length, the size and the weight being thus considerably increased as compared with the high-speed unit. This increase in size with reduction of speed can be noticed in many phases of engineering. Thus compare the size of the motor car or motor cycle engine with that of a gas engine of the same power. A twin-cylinder motor cycle engine of 7 to 8 horse power runs at perhaps 2,000 revs. per min., the gas engine runs at 200 revs. per min.; and there is hardly any comparison between the size of the two, the former being small and compact, while the latter is large and cumbersome. The reason for this remarkable disparity in size is, briefly, that the work done in any engine is the product of force with distance, i.e.

[1] One knot = 6,080 ft. per hour, or 1·15 miles per hour, so that to convert knots into miles per hour multiply by 1·15.

[2] See Volume I, Chapters IV and V.

Work = Force × Distance, and if the speed of rotation of an engine is slow, we must have a larger piston, so that the force on it is greater, the pressure per unit area (i.e. the explosion pressure) being approximately the same in each case, while the force on the piston is equal to the product of the pressure and the area. Moreover, the piston must move through a longer stroke so that the distance it moves through in a given time—stroke of piston × number of strokes—shall be as large as possible. Similarly in the steam turbine, the pressure of the steam will not be greater because the turbine runs at slow speed, but by increasing the diameter of the rotor, the distance moved through during each revolution is also increased and thus the loss of energy due to a reduction of speed is more or less counterbalanced. But unfortunately, from the marine engineer's point of view, the increase of diameter means increase in size and weight, both of which are very disadvantageous on board ship.

Reference has just been made to the motor car engine, which can certainly be described as a high-speed engine. The reader must know that for driving motor cars it is essential that the speed of rotation of the driving shaft shall be reduced between the engine and the back axle, and in order to do this some form of gear is employed, the function of which is to enable the speed of the driving wheel to be brought down to a figure which gives a reasonable road speed to the car. It may be that this gave to Sir Charles Parsons the idea of reducing the speed of the turbine in a similar manner, i.e. by the use of gears. In any case, it was recognised that if the steam turbine with its many advantages was to be made suitable for use on all types of ships and particularly on slow-speed cargo ships—which after all form the most important part of the British mercantile marine, both as regards numbers and total tonnage—then some method of reducing the speed between the turbine and the propeller was very desirable. The problem was not, however, a simple one, since the powers to be dealt with were many times greater than those used on motor cars, where gears were so successfully employed. Eventually, Sir Charles Parsons, to whom the honour of first employing speed reduction gearing on a cargo ship rightly belongs, designed his single-reduction double-helical gearing for the cargo steamer *Vespasian*, a vessel which had previously been thoroughly tested with her original reciprocating engines. The vessel was then fitted with steam turbines driving the propeller through the speed reduction gearing and a further series of trials was carried out in order to compare the performance of the two types of propelling machinery, viz. the reciprocating steam engine and the geared turbine. The results of these comparative trials were made public by Sir Charles Parsons in a paper which he contributed to the Institution of Naval Architects in 1910. We need not go further into the details of this installation, except to note that the trials showed that an economy of over 15 per cent. in coal consumption was obtained by the substitution of geared turbines for the original reciprocating engines.

Following these trials it was decided to build a new vessel specially designed for geared turbine machinery, and the result was the *Cairnross*, which can rightly claim to be the pioneer vessel built for geared turbine propulsion. The particulars of this ship and the results of a series of trials run in conjunction with a sister vessel, the *Cairngowan*, of exactly the same dimensions, propelled by reciprocating engines will also be found in the *Transactions of the Institution of Naval Architects*. Both vessels were 369·75 ft. in length and 50·75 ft. in breadth, with a displacement of approximately 10,000 tons; and as indicating the economy of the geared turbine for marine propulsion, whereas the *Cairngowan* showed a coal consumption of 32·692 tons per day, that of the *Cairnross* was 27·816 tons per day, representing a saving of 14·9 per cent. in favour of the latter vessel. The gearing on the *Cairnross* consisted of a pinion direct coupled to the turbine shaft and driving a gear wheel to which the propeller shaft was coupled. Double helical teeth were cut on both pinion and wheel, there being 26 times as many teeth on the latter as

on the former. Thus the speed was reduced in the ratio of 26 to 1, and the turbine speed of 1,700 revs. per min. was reduced to 65 revs. per min. at the propeller. In this way it was possible to employ an economical high-speed turbine in conjunction with an efficient low-speed propeller, the result being as indicated by the coal consumption given above.

The remarkable results obtained with the *Vespasian* and the *Cairnross* led to the building of a large number of vessels in which gearing was installed in conjunction with steam turbines. Not only on slow-speed cargo vessels, but on high-speed liners and on warships it gradually became apparent that the use of gearing was highly advantageous, since it enabled both the turbine and the propeller to be run at their most economical speed. We find, further, that the tendency was for the speed of the turbine to be increased, and thus a higher reduction ratio was demanded at the gear. When, however, a very small pinion is employed to drive a very large gear wheel, even as in the case of the *Cairnross*, in which the gear wheel has a diameter 26 times as great as that of the pinion, it is apparent that the number of teeth in contact must be small unless the pinion is made of large diameter, in which case the gear wheel becomes enormous. On the other hand, with a small pinion there is a possibility of failure due to the large force transmitted at the point of contact of the teeth, particularly on high-powered vessels. To obviate these difficulties and at the same time to enable even greater speed-reduction ratios to be obtained, the "double-reduction" gear was introduced. In this gear, as the name implies, the reduction between the turbine shaft and the propeller shaft takes place in two steps. The turbine is directly coupled to a double helical toothed pinion, and this drives a gear wheel mounted upon an intermediate shaft, the same shaft also carrying a second double helical toothed pinion and this second pinion drives a second gear wheel mounted upon a third shaft, which is coupled to the propeller shaft. Thus if the speed reduction ratio between each pinion and its gear wheel is 5 to 1, we get a total speed reduction ratio between the turbine and the propeller of 25 to 1. The whole of the gearing is mounted in a single casing, and as there are no excessively large wheels, it does not occupy any considerable space in the engine room. With the double-reduction gear it has been found possible to employ reduction ratios exceeding 50 to 1. Thus in one particular example the speed of the turbine is 4,000 revs. per min., and that of the propeller 77 revs. per min., giving a ratio of 52 to 1. With this occurring in two steps, we get at each gear a reduction ratio of about 7·2 to 1, although it may be noted that it is not necessary that the reduction in both steps should be exactly the same.

The thought may perhaps have occurred to the reader that this reduction of speed must of necessity be accompanied by a certain loss of energy due to friction and noise in the gear. Such, of course, is undoubtedly the case, but the actual loss of power is only a very small percentage of the total transmitted. The design and construction of marine reduction gearing has been carried to a very high degree of perfection, with the result that the losses are very small indeed. It has been found as a result of actual tests that the efficiency of the transmission through a single-reduction gear is as high as 98 per cent., that is, only 2 per cent. of the energy is wasted in friction. This, of course, applies to gearing which has been very carefully cut and in which the lubrication of the bearings and of the teeth themselves is thoroughly efficient. Under such circumstances there is not any possibility of serious loss, since friction can only occur at the very small area at which the teeth are actually in contact, and at the bearings. The former are made of highly polished steel with jets of oil under pressure squirted right up against the point of contact, and the bearings, which are white metal lined, are also lubricated under pressure. In the double-reduction gear where there are two lines of contact and where there are more bearings, the amount of friction is necessarily greater; but even so, the efficiency of a good double-reduction gear should be about 95 per cent. It is of interest to note that while an efficiency of 98 per cent. for a single-reduction gear appears to be very high,

it must not be forgotten that on a ship requiring 2,000 horse power to drive it at the desired speed, this represents a loss of 40 h.p., or, in other words, the power required to drive the gear is 40 h.p. With double-reduction gear the power actually utilised to drive the gear itself is greater, but with both types the performance, as indicated by the percentage efficiency, is extremely creditable.

The question has often been raised as to whether now that gears are employed on board ship it would not be possible to arrange for a reversing gear in addition to the speed reduction gear, thus eliminating the astern turbines. To indicate how impossible this idea is, it need only be stated that the idea has never been tried as far as ocean-going cargo ships are concerned. When one calls to mind the fact that we are dealing now with powers exceeding 1,000 h.p., it will at once be realised that the size of these gears is vastly different from those found even on the most powerful motor car and that the possibility of throwing wheels in and out of mesh with the aid of a lever, even if power were employed for operating the reversing mechanism, is rather remote. On larger vessels where the power transmitted by each gear is considerably in excess of that mentioned above, the question must at once be ruled out as being utterly impracticable. The largest marine speed reduction gears ever constructed are those fitted on H.M.S. *Hood*, each of which transmit 35,000 h.p., there being four shafts, the total power developed by the turbines being 140,000 h.p., and in view of the limitation in the size of naval vessels imposed by the Washington Pact, it is hardly likely that this will be exceeded for a long while to come. Our point is, however, that with the size and weight of the wheels in gears such as these it is obviously impossible to introduce an arrangement for reversing the direction of rotation of the propeller shaft while the turbine still runs in the same direction.

It is a remarkable fact that while single-reduction gears have proved remarkably successful both on merchant ships and warships, a number of failures have been recorded with double-reduction gears. The reason for this is rather difficult to understand, because, as might be expected, great care has been taken in the design to ensure that the various parts shall be made of ample strength and all the parts are machined with the same extreme accuracy. It is generally admitted that these failures have been due to vibrations set up in the propeller shaft (due to the blades of the propeller alternately working through deep and shallow water), which are transmitted to the second gear wheel and thence to the intermediate shaft carrying the second pinion and the first gear wheel. This shaft is, as it were, only floating between the turbine shaft and the propeller shaft, and in consequence it magnifies the vibrations in the propeller shaft with the result that rapid wear, and ultimately failure, occur. By excessive strengthening of the various parts beyond that which is demanded by the actual power transmitted, it has been found possible to reduce these failures to a considerable extent. Moreover, certain devices have been introduced with a view to eliminating the vibration in the different parts of the gear, but the fact remains that the double-reduction gear has not met with so emphatic and so uninterrupted a measure of success as has the single-reduction gear. It is a significant fact that only single-reduction gears are to be found on British naval vessels, although this may be partly accounted for by the very high speeds at which modern warships are propelled.

One result of the lack of success with double-reduction gearing has been the bringing forward of alternative methods for reducing the speed between the turbine and the propeller. Two systems have been tried, viz. the hydraulic and the electric, although the former has at the time of writing hardly progressed beyond the initial stage. In the hydraulic speed reduction system, the turbine is directly coupled to a high-speed pump which in turn delivers water to a low-speed water motor. Arrangements are made for varying the speed of the motor between zero and the maximum, and it is possible to reverse the motor by merely reversing the direction of the flow of the water without in any way

interfering with the running of the turbine. It is difficult to say whether this system is commercially superior to the mechanical gear system, but it may be mentioned that the efficiency of the transmission in the former case is considerably lower than in the latter. In the electric system, the turbine is directly connected to an electric generator, and the current thus produced is used to drive an electric motor coupled to the propeller shaft, variations of speed and reversal of the direction of running of the propeller shaft being obtained by controlling the magnitude and direction of the current to the motors. A further advantage of this system lies in the fact that it is possible to arrange the turbo-generators in the most convenient position with the motors in any other position as far removed from the generators as desired. Thus with cargo ships, the motors can be placed right aft with only the shortest possible shaft connecting them to the propellers. In this way the space occupied by the shaft tunnels connecting the engine room, which is usually placed amidships, to the after part of the vessel, is made available for the carriage of cargo. This advantage is not obtained with the hydraulic system, but the efficiency of the electric system is likewise rather low, certainly not exceeding 90 per cent.

The electrical speed reduction system has been somewhat extensively employed in the United States Navy and it has undoubtedly proved to be extremely satisfactory for nearly all classes of naval vessels. But while the British naval authorities have very carefully considered the possibility of employing the system on the vessels of His Majesty's Navy, they have come to the conclusion that the change would be inadvisable. Electric propulsion is, it may be mentioned, particularly suitable for naval vessels where frequent changes of speed are required and where the ship may steam for long periods at cruising speed. On merchant vessels the engines are generally run throughout each voyage at about the maximum economical speed for which they were designed. Again, during war practice and in actual war service the power to manœuvre rapidly, going either forward or astern at various speeds, is of great value to a naval vessel, whereas, except when entering or leaving port, manœuvring power is not demanded of a merchant ship. Thus electric propulsion possesses greater attractions to the naval engineer than it does to his brother in the mercantile marine. In spite of this, however, we do find that a few merchant ships have been equipped for electrical propulsion, but time alone will prove whether these may be considered commercially successful.

An even more recent development in marine engineering than the use of the steam turbine has been the application of the heavy-oil engine to ship propulsion. On cargo vessels where it is desirable that the greatest possible space should be available for the actual stowage of cargo, it is obvious that the saving of space due to the elimination of the boilers and the coal bunkers—oil fuel being generally carried in the double bottom of a ship—would be of considerable value. Again, the reduced fuel consumption of the heavy-oil engine, as compared with the steam engine using coal—one ton of oil being equivalent to about four tons of coal—means that a considerably smaller weight of fuel is required for any particular voyage. These and certain other advantages have led to the development of the motor ship, in which Diesel engines using heavy oil as fuel are installed, the engines being designed to run at low speed and direct coupled to the propeller shaft. The Diesel engine suffers, however, from very serious limitations, due to the very high temperatures which occur in the cylinders, and thus it is found very difficult to construct a cylinder of a diameter greater than about 30 in. and the power developed per cylinder is limited to about 500. Thus, where large powers are concerned it is necessary to employ two or more engines, each consisting of six or eight cylinders, a procedure which renders the first cost of the vessel exceedingly high. Attention may here be drawn to the eight-cylinder engine described and illustrated in Chapter II of the first volume of this work as indicating the type of heavy-oil engine used for marine propulsion. This engine, which has cylinders 29·8 in. diam., is operated on the four-stroke cycle and develops

3,000 h.p. at a speed of 115 revs. per min., which represents 375 h.p. per cylinder. For the largest vessels, such as fast passenger liners, the power requirements are such that it has not yet been found possible to propel these with heavy-oil engines and to-day the geared steam turbine is invariably employed. But for cargo ships of moderate dimensions the heavy-oil engine is gradually becoming recognised as one of the most suitable forms of prime mover that is available, and in consequence, far more attention has been paid to its development for marine purposes than for land purposes. In fact it can be claimed that the heavy-oil engine furnishes an example of an engine being primarily developed for marine purposes, and as a result of the experience gained in this sphere, the engine has been subsequently developed for land purposes along the lines which marine practice has shown to be correct.

In the earlier Diesel ships, the main engines were generally operated on the four-stroke cycle, two, four, or six cylinder units being installed. In this way not only was it possible to keep the power developed per cylinder very low, but having two engines there was less possibility of the ship becoming totally disabled due to a breakdown in the machinery. The marine heavy-oil engine is only just beginning to outlive its reputation for unreliability, and it was always considered advisable to have two engines in the hope that they would not both become disabled at the same time, so that, in the event of failure of one engine, the vessel would always be able to limp along on one propeller while repairs were being effected or until port was reached. With the experience that has been gained with the pioneer motor ships, and by considering carefully the causes of these early failures, it has gradually become possible to produce an engine of greater and greater reliability, until to-day it is often claimed that the marine Diesel is equal to the steam turbine in this respect. Thus we find that single-screw motor ships are now being built, the whole power for driving the vessel being obtained from a single engine of four or six cylinders. Further, the tendency has been all in the direction of increasing the power developed per cylinder, or in other words reducing the total number of cylinders required, which in turn means the reduction in the number of working parts where possible failure might occur. It has, therefore, become possible to install Diesel engines on vessels of increasing size, and not only large cargo vessels requiring perhaps 3,000 h.p. to propel them at the requisite speed, but even liners of the intermediate type—i.e. combined cargo and passenger vessels—are now being fitted with this type of machinery.

Concurrent with this increase in the power developed per cylinder has been the gradual introduction of the two-stroke engine. We have seen in that part of our work dealing with the heavy-oil engine[1] that for a given size of cylinder the power developed by the two-stroke engine is almost twice as much as that in the four-stroke engine, but unfortunately the average temperature in the cylinder is far higher, since the combustion occurs twice as frequently in the former than it does in the latter, and thus the liability to fail is greater, since the majority of failures may be said to be due to the effects of high temperature. Reliability is, of course, one of the most vital requirements of a marine engine, being certainly far more vital for a marine engine than for a land engine. With the former, failure may occur when the vessel is thousands of miles from any port and the engineers are left to their own meagre resources to effect repairs, whereas with the latter, there are completely equipped works almost immediately available and new parts can be quickly obtained. For these reasons the two-stroke engine did not at first make any considerable headway, although as it slowly proved its reliability it was gradually adopted because of its inherent advantages over its rival the four-stroke engine, until to-day the position is that about 50 per cent. of the heavy-oil engines built for marine purposes are operated on the two-stroke cycle, and instances can be recorded

[1] See Volume II, Chapter VI.

of makers who have given up the construction of the four-stroke type in favour of the two-stroke type. On the other hand, many marine engineering firms of the highest repute still maintain that the four-stroke engine is the better of the two in view of its greater reliability. In other words, the " battle of the cycles," as it has been termed, has not yet been brought to a decisive issue.

Among other developments, mention may be made of the opposed piston type of engine. The natural sequence of the endeavour to increase the power per cylinder would appear to have been the adoption of the double-acting principle in which power is developed on both sides of the piston. The limitations to which we have already referred have so far rendered this impracticable, although designers are still grappling with the problem of the double-acting heavy-oil engine and one or two experimental engines have been built. As an alternative, the opposed piston engine has been devised in which two pistons are arranged in each cylinder, the combustion of the fuel occurring between them. The pistons thus move outwards in opposite directions, and by a multiplicity of cranks or by cross-connecting the opposite pistons in adjacent cylinders the motion of the pistons is transmitted to the crank shaft. This has enabled considerably greater powers to be developed per cylinder, and in one notable engine, viz. the Doxford opposed piston engine, the power developed by a four-cylinder unit is 3,000 h.p. or 750 h.p. per cylinder.[1] Mention must be made of the Still engine, which has also been adapted for marine work. In this engine the double-acting principle is obtained by employing the internal combustion cycle using heavy oil on the top side of the piston and the steam cycle on the underside, the steam being obtained from the heat of the exhaust supplemented if necessary by a small boiler using the same oil as fuel.

In nearly every form of marine Diesel engine it has been found almost impossible to bring the speed of rotation down to that which is required to enable the most economical propeller to be adopted. We have referred in the earlier part of this chapter to the question of economical propeller speed, and while the speed of the heavy-oil engine does not differ so very greatly from that required at the propeller, the fact remains that they are not quite in agreement. Thus it has not been found possible to bring down the speed of the Diesel engine—either four-stroke or two-stroke engine—to much below 100 revs. per min., and it would be highly advantageous, as enabling a lighter and smaller engine to be built for a given power, if higher speeds could be adopted. On the other hand, for the majority of cargo vessels the economical speed of which is 10 to 12 knots, a propeller speed of about 70 revs. per min. is required. It has, therefore, been suggested that some form of gearing should be introduced between the engine and the propeller. Mechanical gearing offers the simplest solution of the problem, a gear having a reduction ratio of 5 or 6 to 1 being interposed between an engine running at between 300 and 400 revs. per min. and the propeller shaft. Unfortunately, however, the turning effort on the crank shaft of a Diesel engine and indeed on any reciprocating engine is not nearly so even as on the shaft of a turbine, and hence the vibration forces occur similar to those which have been mentioned as leading to failure with double-reduction gearing or turbine-driven vessels. To overcome the effect of these would demand very heavy gearing, and so far the idea has not been developed except in one or two isolated cases and then for special reasons. The electrical drive offers a more satisfactory alternative since it also enables the troublesome reversing gear to be eliminated from the Diesel engine. The main engines, running at their most economical speed, drive electrical generators, and the current from these, drives the slow-speed motors which are direct coupled to the propeller shaft, all the manœuvring and reversing being carried out on the motors. This system has already been tried and more may perhaps be heard of it in the future; but the extra

[1] See Volume II, Chapter IX, for a description of the different types of opposed piston engines, and also of the Still engine.

cost of the electrical machinery hardly enables it to compete on a commercial basis with the direct Diesel drive.

In the foregoing résumé of the recent developments in the science and practice of ship propulsion, we have not considered it necessary for us to describe in detail the various forms of engines referred to. In the first volume of this work dealing with the steam turbine, we had occasion more than once to describe certain specific examples of steam turbines designed and constructed for use on board ship, and we may therefore assume that the reader is now familiar with the general principles underlying the design and construction of that type of prime mover. He is, however, again referred to that section of our work if he is desirous of understanding more about the peculiarities of the marine turbine. Similarly, in our second volume we dealt very fully with the internal combustion engine, and in describing the various forms of heavy-oil engine we were compelled frequently to deal with actual examples of marine oil engines, since the development of this form of engine has undoubtedly been largely due to its enormous possibilities as a prime mover for cargo vessels. Thus it is unnecessary for us to describe again the various forms of heavy-oil engine, and the reader is referred back to Volume II for details of oil engines designed specifically for use on board ship, as well as for descriptions of the opposed piston engines and the Still engine.

There still remains, however, a very important form of marine engine with which, so far, we have not dealt, and that is the reciprocating steam engine. Exactly how important this is can be gathered from the figures in *Lloyd's Register, Annual Survey of the World's Shipping*, which year by year gives a summary of the sizes of vessels built all over the world and, among other information, the type of propelling machinery installed. Thus in the returns published in 1923, neglecting war vessels, small vessels of less than 100 tons gross, and all vessels propelled by sails—the total of these latter was, it is interesting to note, 107 vessels of an aggregate of 41,355 tons—we find that of the total of 745 vessels built during the past year aggregating 2,425,729 tons, 103 vessels of 209,557 tons gross were propelled by oil engines, while of the 642 steamers representing 2,216,172 tons, 104 vessels with an aggregate of about 776,000 tons were fitted with steam turbines. Thus there were no less than 538 vessels of a total of 1,440,172 tons gross propelled by reciprocating steam engines, which represents over 72 per cent. of the total number of the mechanically propelled vessels launched during the year. One other significant fact may be noted, viz. the average tonnage of the vessels fitted with the different types of propelling machinery. It will be found from the above figures that the average size of the 103 motor ships was 2,020 tons, that of the 104 steam turbine propelled vessels was about 7,460 tons, while the average gross tonnage of the 538 vessels propelled by steam reciprocating engines was 2,680. It is beyond the scope of this book to look further into the significance of these figures, but the broad conclusion may be noted, viz. that the larger vessels of the passenger-liner class are being built for steam turbine propulsion, while for the smaller vessels of the cargo-carrying type the steam reciprocating engine is used in preponderating numbers, although there is a small proportion which is fitted with internal combustion engines, this proportion showing signs of a slow but steady increase.

While the statistics published in the annual survey issued by *Lloyd's Register* are silent upon this point—they could, of course, be determined by a careful study of the particulars published in *Lloyd's Register Book*—it is safe to say that of the 538 vessels propelled by steam reciprocating engines, by far the larger majority were propelled by a single vertical triple-expansion engine. This type of engine can be regarded almost as the standard form for the propulsion of cargo steamers, although, as has already been noted, there is a tendency for the steam turbine and the heavy-oil engine of the Diesel type, particularly the latter, to oust it from the proud position which it once held. Where

larger powers are required and the shipowner still desires to keep to the well-tried reciprocating engine for the propulsion of his vessel, then the quadruple-expansion type is employed with twin screws. A few of these engines are included in the above figures, although for larger ships of the liner class the steam turbine now holds almost undisputed sway. The reader may well ask why, in view of the undoubted economy of the steam turbine and particularly of the heavy-oil internal combustion engine, the steam reciprocating engine is still employed? The answer is to be found in the one word "reliability." It is apparent that with a vessel at sea carrying a cargo worth perhaps a hundred thousand pounds or more, the risk of breakdown of the main propelling machinery must be carefully guarded against, and to many shipowners the steam turbine as available for moderate

FIG. 128.—A Typical Cargo Steamer, s.s. *Northwestern Miller*, built by the Northumberland Shipbuilding Co., Ltd., and engined by Messrs. Richardsons, Westgarth & Co., Ltd.

powers and the heavy-oil engine are not sufficiently reliable, and they are prepared to purchase reliability at the expense of higher fuel costs. It is not for us to discuss here the wisdom of their decision.

In view, then, of the very great importance of the steam reciprocating engine, it it necessary for us to describe and illustrate one or two typical installations. As an example of the triple-expansion engine which is to be found on the bulk of our cargo vessels, we may consider the engines of the s.s. *Northwestern Miller*. This vessel was built by the Northumberland Shipbuilding Co., Ltd., at Howdon-on-Tyne, for the Philadelphia Transatlantic Line, owned by the well-known shipping firm Furness, Withy & Co., Ltd., London, and the engines were designed and constructed by Messrs. Richardsons, Westgarth & Co., Ltd., a very prominent marine engineering firm having works at Hartlepool, Sunderland, and Middlesbrough. We are indebted to Messrs. Richardsons, Westgarth & Co., Ltd., for the photograph of the *Northwestern Miller*, reproduced in

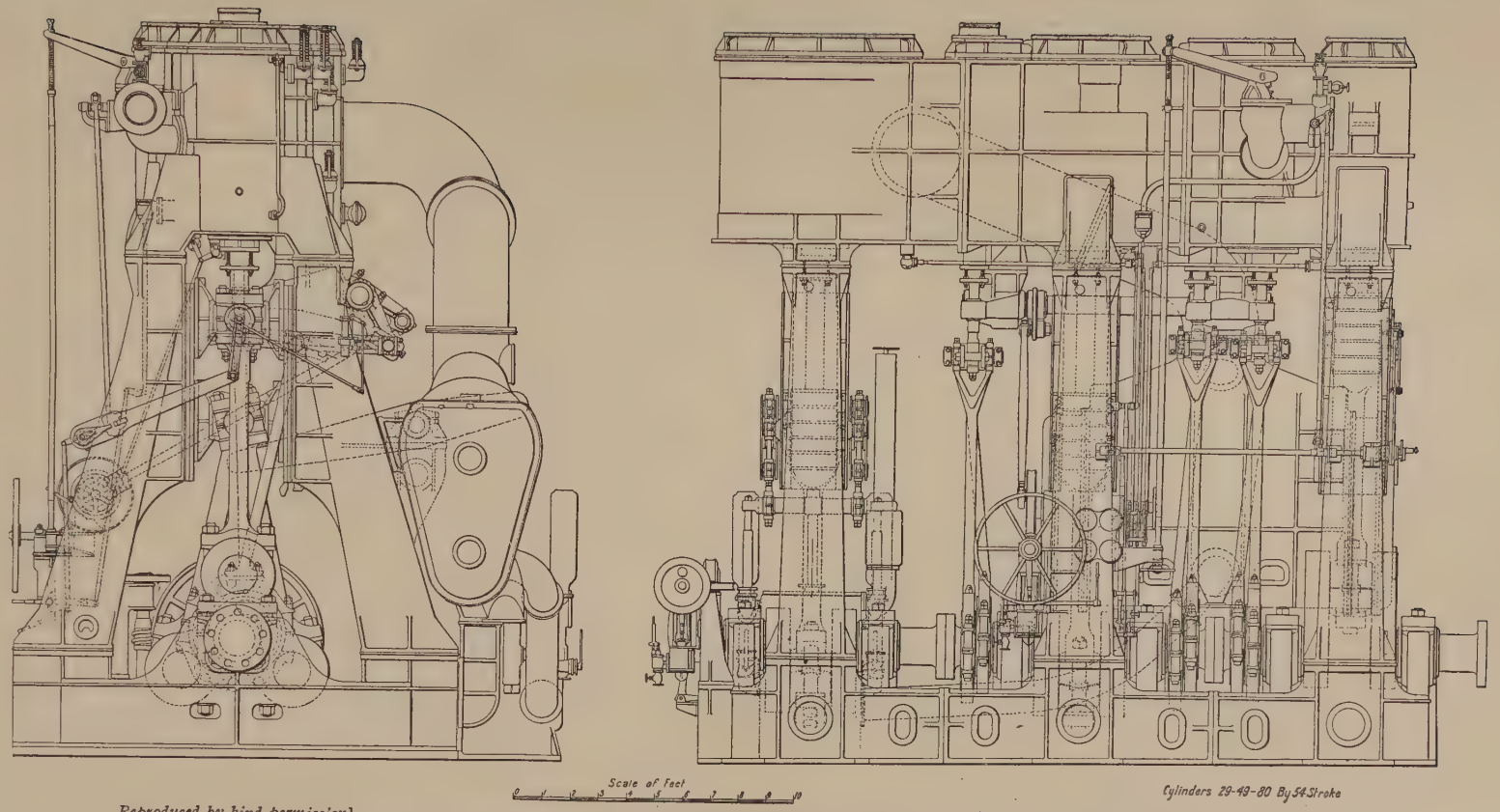


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FIG. 129.—General Arrangement Drawings of the Triple-expansion Engines of s.s. *Northwestern Miller*, designed and constructed by Messrs. Richardsons, Westgarth & Co., Ltd.

Fig. 128, from which it will be noted that the vessel is of the ordinary cargo-carrying type with the propelling machinery placed amidships, practically the whole of the hull being available for the carriage of cargo, for which the total space available is 538,500 cu. ft. Except for a small distance on either side of the funnel, where the navigating bridge, the chart-house, and the quarters for the captain, the officers, and the engineers

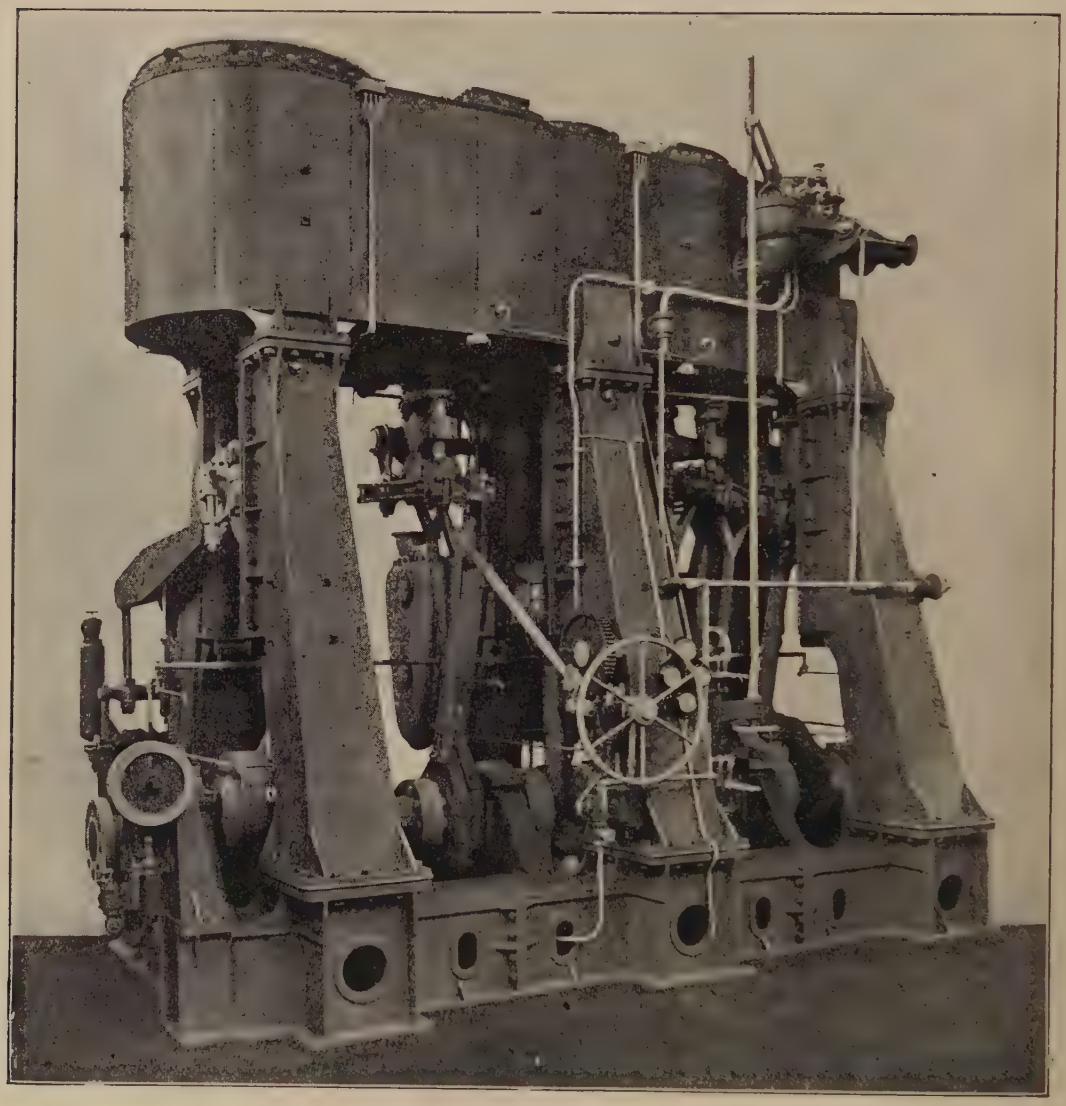

Fig. 130.—The Triple expansion Engines of s.s. *Northwestern Miller*. Front View, showing Stop Valve and Reversing Gear Controls.

are situated, the entire deck is clear, being occupied by the hatchways and the winches, which are employed for working the cargo in and out of the holds. The cargo derricks are shown in the "stowed" position, i.e. the position they occupy when the vessel is at sea.

We are not, however, concerned so much with the vessel herself as with her propelling machinery. The vessel is propelled by a single screw, which is direct coupled to a triple-expansion engine by a long shaft situated in a tunnel at the bottom of the

after-holds through which the engineers can pass in order to examine the shaft bearings and the stern gland, i.e. the gland through which the shaft passes out of the vessel to the propeller. The principal features of the engine will be readily understood from a study of the drawing reproduced in Fig. 129, for which we are indebted to the editor of *Shipbuilding and Shipping Record*, in which journal the drawings and particulars of this

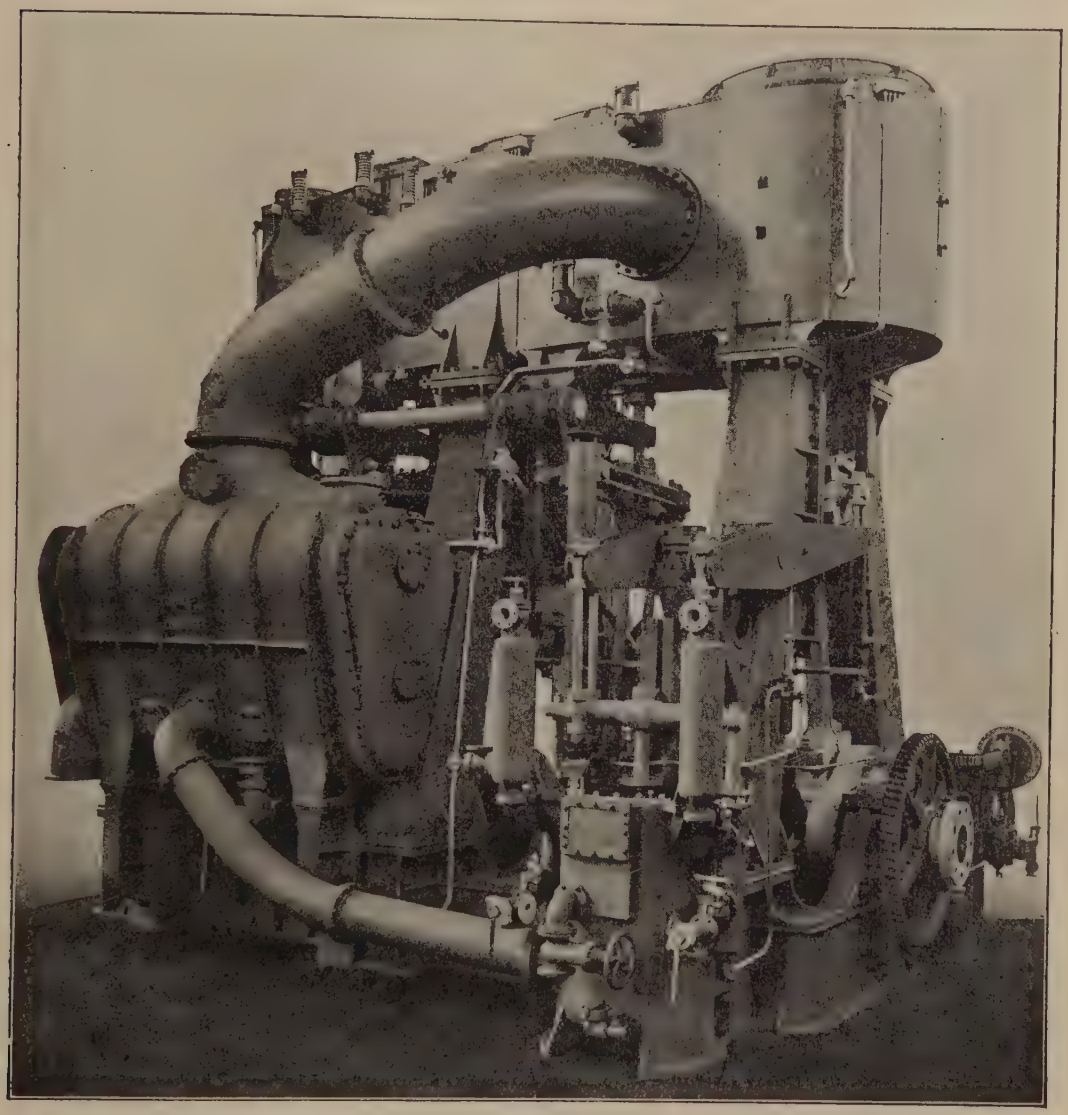

FIG. 131.—The Triple-expansion Engines of s.s. *Northwestern Miller*. Back View, showing Condenser, Air Pump, and Circulating Pump.

vessel and her propelling machinery were first published. The appearance of these engines as erected in the builders' shops will be gathered from Figs. 130 and 131, which show respectively the front view with the main controls and the back view with the condenser air pump and circulating pump. The high-pressure cylinder is 29 in. diameter, the intermediate-pressure cylinder is 49 in. diameter, while the diameter of the low-pressure cylinder is 80 in., the stroke of the piston in each cylinder being 54 inches. When supplied with steam at a pressure of 180 lb. per sq. in., the engine

develops 3,250 indicated horse power at a speed of rotation of about 75 revs. per min. As will be seen from Fig. 130, the valve gear is operated through Stephenson's link motion, and in order to facilitate the work of manœuvring, the reversing gear is operated by a small steam engine, so that the links can be readily moved from full ahead to full astern or stopped in any intermediate position as required. A large hand wheel is also provided for operating the reversing gear, in case the small steam engine is out of action. The main steam stop valve is operated by means of a smaller hand wheel mounted on the other side of the column, provision being made for by-passing high-pressure steam to the intermediate and low-pressure cylinders during starting up or manœuvring should the engine stop with the high-pressure crank on either of its dead centres. The by-pass valves are operated by the small hand levers seen just below the stop valve wheel. A small engine is fitted for turning the main engine when the vessel is in port so that it does not get fixed in any one position. The condenser, which is of the contraflo pear-shaped type, is mounted on the columns at the back of the engine (see Fig. 131). It has a total of 4,000 sq. ft. of cooling surface. For dealing with the condensate and with the air and other incondensable vapour, an air pump of the Edwards pattern is fitted,[1] driven through working levers from the low-pressure crosshead. No special device is fitted for dealing separately with the, air but the size of the air pump and of the circulating pump is such that a vacuum of 27½ in. can be maintained in the condenser under normal conditions. Steam for driving the main engine as well as for operating the various auxiliaries, such as the dynamo engine, the feed pumps, the bilge and ballast pumps, the deck winches, etc., etc., is supplied by three boilers of the usual return tube cylindrical marine type,[2] each 17 ft. 6 in. diameter and 12 ft. long, having a working pressure of 180 lb. per sq. in. and yielding a total heating surface of 10,450 sq. ft. The boilers are coal fired and the products of combustion are arranged to pass up to a single chimney. Particular attention was paid to the design of every detail of the engines and boilers in order that they might withstand the severe conditions of the North Atlantic trade, but generally speaking the equipment can be considered as typical of the average cargo vessel.

As an example of the quadruple-expansion engine as fitted to passenger liners we may consider those fitted to the Cunard liner *Laconia*. This vessel was built in 1911, but was lost during the Great War. Another vessel bearing the same name, propelled by double reduction geared turbines, was built in 1921, but we do not hesitate to include in this chapter particulars of the earlier engines, as they indicate the type of engine that was very widely adopted in the days immediately before the steam turbine was deemed suitable for the propulsion of vessels of moderate speed. The propelling machinery of the old *Laconia*—and also that of the new vessel—was designed and constructed by the Wallsend Slipway & Engineering Co., Ltd., of Newcastle-on-Tyne, and consisted of two sets of quadruple-expansion engines driving twin screws. The two sets of engines as erected in the shops just prior to being fitted in the vessel are shown in Fig. 132. In the foreground is the thrust block of one of the engines, i.e. the arrangement for transmitting the main thrust from the propeller to the hull of the vessel. It will be noticed that there were three platforms around the engine, in addition to the engine room floor, which was at the level of the top of the blocks shown, the first giving access to the link motion and the crossheads, the second to the stop valves, etc., and the third to the cylinder tops, while the engine room floor served as the driving platform and gave access to the main bearings, the eccentrics, etc. The relative size of the men leaning beside the blocks gives a good idea of the general overall dimensions of these engines.

[1] See Volume I, Chapter VIII, for a description of this type of pump.

[2] See Volume IV, Chapter I, for a description of the usual type of marine boiler, as well as for the results of trials of triple-expansion marine engines.

The engines were designed to run at a speed of 89 revs. per min. when supplied with steam (not superheated) at a pressure of 210 lb. per sq. in., and each engine developed 7,250 i.h.p. The vessel was 600 ft. long and the total of 14,500 i.h.p. was sufficient to give

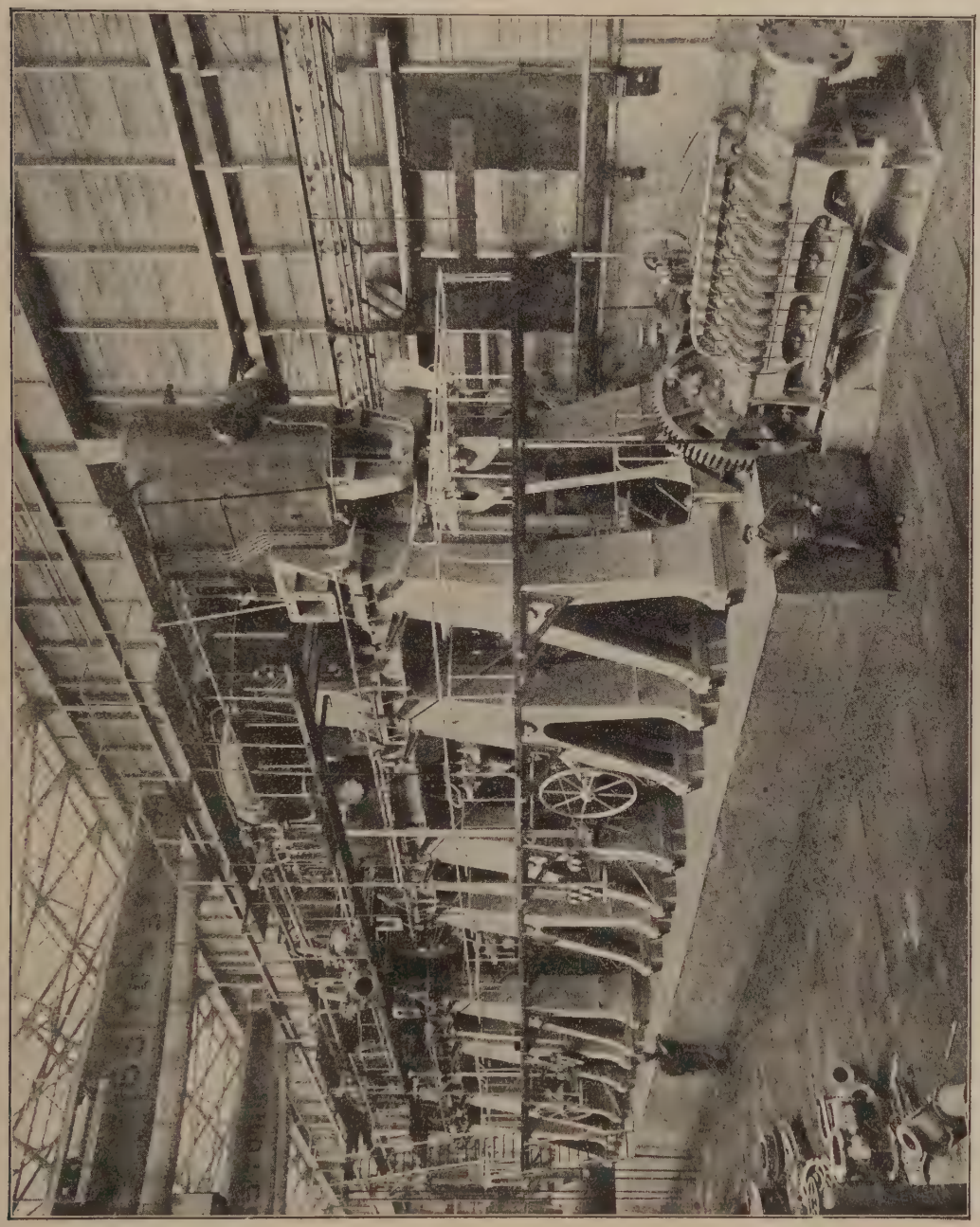

FIG. 132.—Two Sets of Quadruple-expansion Engines by the Wallsend Slipway & Engineering Co., Ltd., each developing 7,250 i.h.p.

her a speed of 16 knots. The four cranks of each engine were not arranged quite at 90° to each other, but at angles differing slightly from this in order to give as perfect a balance as possible and to reduce vibration to a minimum. The high-pressure cylinder was 33 in. diameter, the first intermediate-pressure cylinder was 47 in., the second

intermediate was 67 in., and the low-pressure cylinder was 95 in. diameter, each of them having a piston stroke of 60 in. Each cylinder was fitted with a separate liner, and the second intermediate and the low-pressure cylinder, i.e. the two in which the steam was at its lowest temperature, were steam-jacketed. Piston valves were fitted to the high and to the first and second intermediate-pressure cylinders and slide valves to the low-pressure cylinders. The valve gear, which was of the Stephenson's link type, was worked by means of the usual double eccentrics and link motion, the high-pressure gear being seen in the foreground of the picture. Reversing was accomplished by means of a combined steam and hydraulic engine, and all the control gear was arranged conveniently at the centre of the engine as shown. The working parts of the engine were made of forged steel and the shafting was made by Sir W. G. Armstrong, Whitworth & Co., Ltd., of hydraulic forged fluid compressed steel. The propellers were built up with four manganese bronze blades.

The condensers were arranged independently of the framing of the main engines, but were erected very close to them in order to economise space in the engine room. They were of Weir's "Uniflux" type built up of steel plates, and they were operated in conjunction with air pumps of the "Dual" type supplied by G. & J. Weir, Ltd. The circulating pumps were of the centrifugal pattern driven by independent engines, in fact no pumps whatever were driven from the main engines. There were three main feed pumps of Weir's automatic duplex type, and in addition there was a complete installation of pumps for dealing with the bilge, the ballast, sanitary services, etc. For feed make-up purposes and also for providing an ample supply of drinking water, four evaporators together with four distilling condensers were carried. There were six boilers of the double-ended cylindrical marine type, each 17 ft. 6 in. mean diameter and 21 ft. 6 in. long, designed for a working pressure of 210 lb. per sq. in., working under natural draught. These were also constructed by the Wallsend Slipway & Engineering Co., Ltd., and were arranged in the vessel in two boiler-rooms, three being placed abreast. Two large funnels were fitted, these being of oval section, 14 ft. 3 in. by 9 ft. 3 in., and having outer casings with air space between. The height of the top of the funnels above the level of the firebars was no less than 139 ft., which gives a good idea of the size of the vessel. This equipment may be considered as typical of that which was fitted in large liners before the reliability of the geared turbine was completely proved. To-day, where engines of that power are required, the geared turbine would generally be fitted.

It must not be thought, however, that, in view of the development of the marine turbine, the quadruple-expansion engine is not being built at the present time. For vessels of the intermediate type, i.e. carrying both passengers and cargo, for which high speeds are not demanded and in which therefore the total power required to drive the vessel is not excessive, the quadruple-expansion engine represents a very economical and reliable form of propelling unit. There has just been completed for the Peninsular and Oriental Steam Navigation Co., Ltd., the famous P. & O. Line—a fleet of five vessels, each 520 ft. long and of about 13,100 tons, for the company's branch line service from the United Kingdom to Australia *via* the Cape. These vessels are propelled by two sets of quadruple-expansion engines, each developing 4,000 i.h.p. at a speed of 95 revs. per min. The following particulars of the equipment of the t.s.s. *Ballarat*, which was the second of the five vessels to be completed, will give a good idea of modern practice with quadruple-expansion marine engines. The *Ballarat* was built by Messrs. Harland & Wolff, Ltd., at their Greenock Yard, and the engines were built by J. G. Kincaid & Co., Ltd., also of Greenock. The diameters of the cylinders are 23½ in., 34½ in., 48½ in., and 70 in. respectively, with a piston stroke of 54 in., the cylinders being arranged in the following order from forward to aft, viz. high pressure, low pressure, second intermediate pressure, and first intermediate pressure. The engines have been very carefully balanced, the

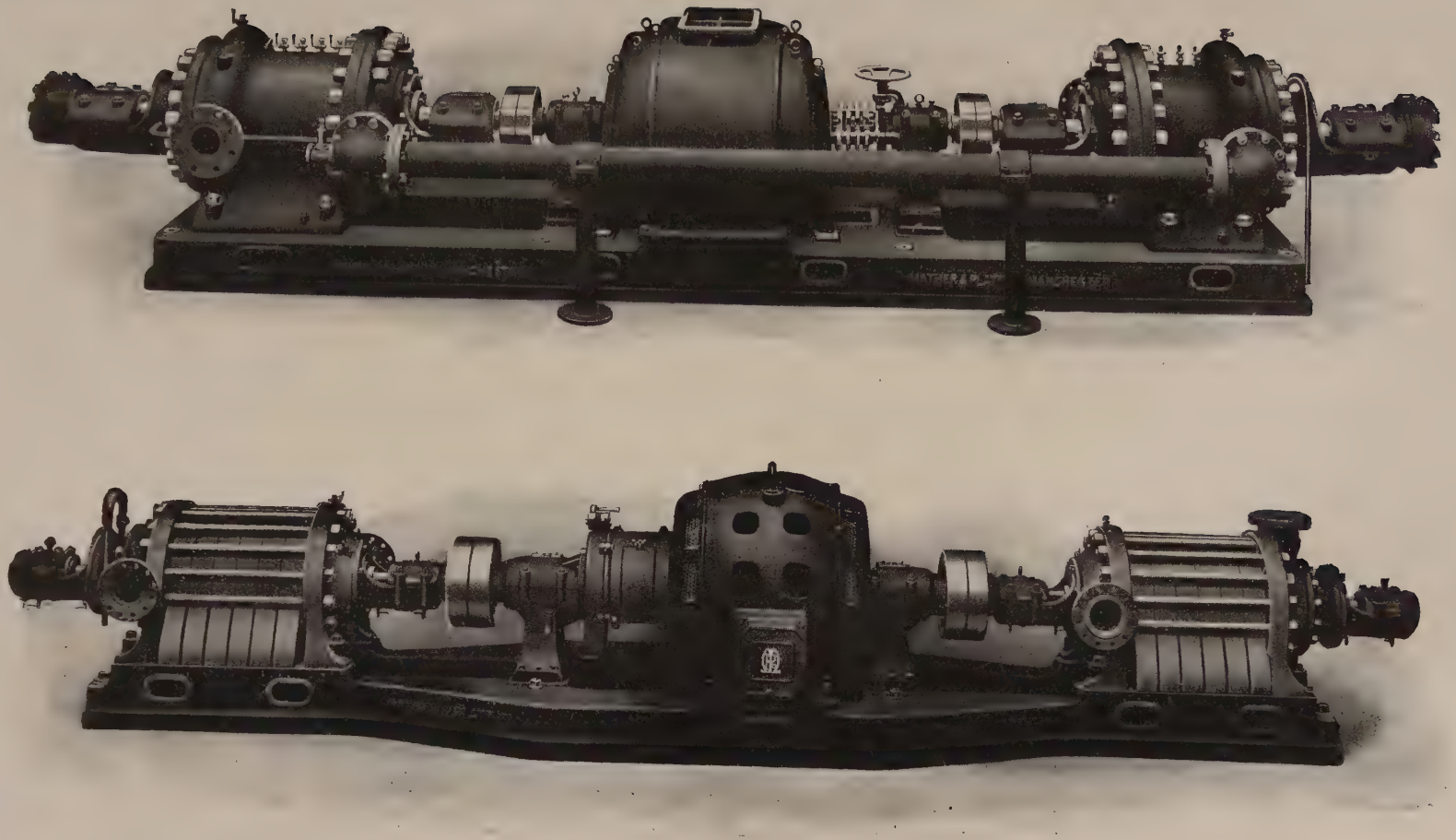

PLATE IV.—TANDEM MULTI-STAGE HIGH-LIFT CENTRIFUGAL PUMPS BY MESSRS. MATHER & PLATT, LTD., MANCHESTER. THE UPPER VIEW SHOWS A PUMP DELIVERING 800 GALLONS OF WATER PER MIN. AT A PRESSURE OF 1,050 LB. PER SQ. IN. THE LOWER VIEW SHOWS A PUMP DELIVERING 800 GALLONS OF WATER PER MIN. AT A PRESSURE OF 750 LB. PER SQ. IN.

cranks being set slightly out of the right-angle position as on the old *Laconia*. The high-pressure and the first intermediate-pressure cylinders are fitted with piston valves, while the remaining cylinders have balanced slide valves. These are all operated by Stephenson's link motion, the link being of the double bar type which is usual in marine practice. The reversing engine is of the combined steam and hydraulic type to which attention has already been drawn, and in addition to the usual arrangement of stop valves and by-pass valves a governor is fitted which comes into operation cutting down the supply of steam if the speed of the engine becomes excessive due, say, to the vessel pitching in a seaway and lifting the propeller out of the water. As soon as the speed falls, steam is admitted again. Thus the governor can only be considered as an overspeed device, and not, as in land engines, an arrangement for keeping the speed of the engine constant at all loads. Such governors are not fitted on marine engines, the speed being controlled from the stop valve to meet requirements which may vary considerably on different voyages owing to change of load, trim, etc.

The main condensers are of Weir's "Uniflux" type, having 4,000 sq. ft. of cooling surface, the air pumps being of Weir's "Dual" type, while the circulating pumps are of the centrifugal pattern driven by high-speed compound vertical engines. The condensing plant is arranged to give a vacuum of 26 in. in the condenser with circulating water at a temperature of 82° F., this comparatively low vacuum being, of course, exceeded when the temperature of the cooling water is lower than 82° F., which represents approximately the temperature of the sea water in the tropics through which the vessel passes. The remainder of the auxiliaries call for no particular notice except the electric generating installation, there being four dynamos each of 50 kw. capacity driven by high-speed vertical steam engines. These supply over 1,200 electric lamps, 33 electric motors, in addition to the motors for 13 electrically operated water-tight doors, which are fitted in various parts of the vessel. This somewhat large amount of light and power is called for to meet the requirements of the 1,235 passengers and the 285 officers and crew which is the vessel's full complement. In addition to the above there is an emergency dynamo of 25 kw. capacity installed on one of the upper decks, so that in the event of the engine room being flooded, power is available for operating the wireless telegraphy and certain emergency pumps as well as for the essential navigating lights. Two large refrigerating machines are installed, since a number of the holds, having a total capacity of 345,000 cu. ft., are arranged for the carriage of frozen produce from Australia.

Steam for driving the main engines as well as for the auxiliaries is supplied by five boilers of the usual marine type, two being double ended and three single ended, all the boilers being arranged in one boiler room. They are all 16 ft. 6 in. diameter, the single ended being 11 ft. 8 in. long and the double ended 20 ft. long, while the total heating surface is 21,320 sq. ft. and the working pressure is 215 lb. per sq. in. The furnaces are coal fired, the vessel having a total bunker capacity of 4,350 tons. It may be of interest to note that on the trial trips when the vessel was in a comparatively light condition drawing only about 20 ft. of water as against the 27 ft. 6 in. full load draught, a speed of 14·52 knots was obtained at a propeller speed of 90·5 revs. per min.; the total power developed by the two engines being 6,355 i.h.p., which compares with a service speed of 14 knots when the vessel is fully loaded. Fig. 133, for which we are indebted to the editor of *The Marine Engineer and Naval Architect*, shows a complete set of indicator diagrams taken during these trials, and a considerable amount of interesting data concerning the running of these engines is given, including the power developed in each cylinder.

Reference has already been made to the four-cylinder triple-expansion type of engine, a type which is still used for marine purposes where large powers are required, although, like the quadruple-expansion, it is seldom installed on new tonnage. The best-known example of the four-cylinder triple-expansion engine is to be found in the famous White Star liner

Olympic. This vessel is propelled by three screws, there being two reciprocating engines on the wing shafts, both of which exhaust into a single low-pressure turbine, which drives the centre shaft. In an early chapter of the first volume of this work,[1] a brief reference was made to the machinery of this vessel and an illustration was given of the reciprocating engines as erected in the shops of Messrs. Harland & Wolff at Belfast, the builders of the vessel and her machinery. A few particulars of these engines may, however, prove of interest. The reciprocating engines each develop 16,000 h.p. at a normal running speed of 77 revs. per min. The high-pressure cylinder is 54 in. diameter, the intermediate is 84 in. diameter, and the two low-pressure cylinders are each 97 in. diameter, the stroke of the piston in each cylinder being 75 in. Steam is supplied by no fewer than 29 boilers, 24 of these being of the double-ended type and five of the single-ended type, all the boilers being 15 ft. 9 in. in mean diameter, while the length of the double-ended boilers is 20 ft. and that of the single-ended boilers 11 ft. 9 in., the working pressure in all the boilers being 215 lb. per sq. in. The reciprocating engines are connected to a hollow shaft which has an external diameter 26¼ in. and an internal diameter 12 in., at the end of which is a three-bladed propeller 23 ft. 9 in. diameter. The exhaust turbine, which develops 18,000 h.p. at a speed of 170 revs. per min., receives its steam from the low-pressure cylinders of the reciprocating engines at a pressure of about 10 lb. per sq. in.[2] It is connected to a hollow shaft 20½ in. external diameter and 10 in. internal diameter, at the end of which is a four-bladed propeller 16 ft. 6 in. diameter. The combined power of the reciprocating engines and turbines, aggregating 50,000 h.p., gives the vessel, which is nearly 900 ft. long and of 46,400 tons gross, a speed of 22½ knots.

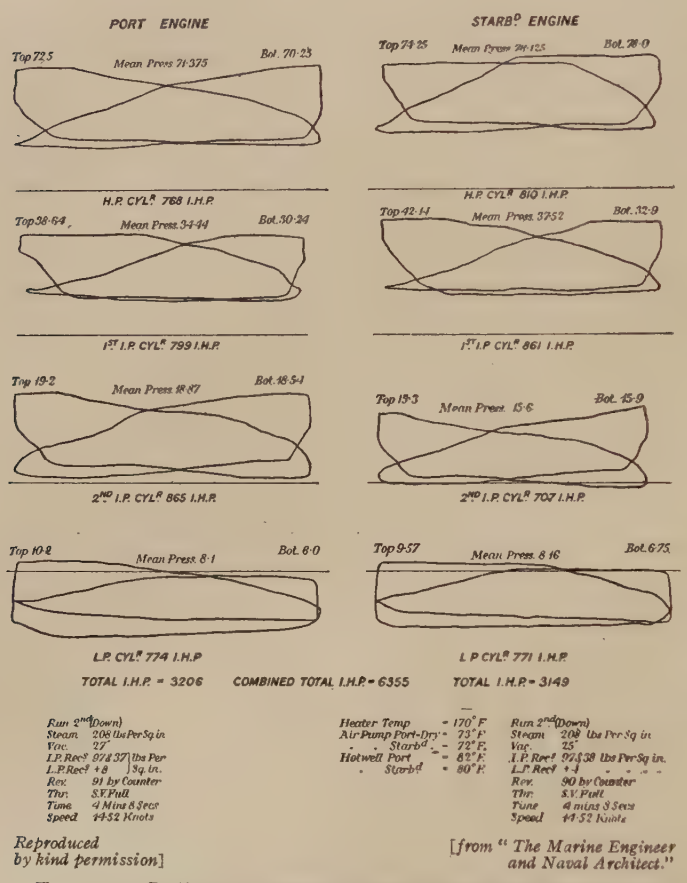

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FIG. 133.—Indicator Diagrams taken during the Trials of the Quadruple-expansion Engines on the t.s.s. *Ballarat*.

It would be possible to deal at considerable length with the subject of the propelling machinery of a ship, but the reader will recognise that marine engineering is of such great importance, particularly to this country, that it deservedly forms a separate branch of engineering, and as such possesses a vast literature of its own to which the reader must be referred if he is desirous of learning more about the special requirements of a

[1] See Volume I, Chapter II.

[2] See Volume I, Chapter VI, for a description of this turbine.

ship's propelling machinery and how they are met. We have endeavoured above to describe some of the typical forms of reciprocating steam engines that are to be met with in the merchant service, so that a comparison can be made with similar engines as used on land. We have in other parts of this work described steam turbines as designed for use on board ship and also heavy-oil engines for the same purpose, and the earnest student should now be able to recognise the essential features of a marine engine, viz. its power to manœuvre, its exceptional reliability and also its comparatively low speed. Another feature is the auxiliary equipment, the whole machinery having to be entirely self-contained in the comparatively small space available in the engine room. To fully understand just how complete this is, it would be necessary to describe in detail the equipment of a vessel such as, say, the *Mauretania* with turbines developing 74,000 h.p. arranged on four shafts, where, in addition to the requirements of engines and boilers, the needs of a small floating town of over 3,000 persons have to be met. It is, however, beyond the scope of this book to deal with this interesting subject, but we may repeat in conclusion that apart from its own intrinsic importance, marine engineering is of the greatest importance because of the effect which it has on the development of almost all branches of land engineering, and for that reason alone, if for no other, it is worthy of being carefully studied.

CHAPTER VI

THE STEAM LOCOMOTIVE—HOW IT WORKS—SOME MODERN EXAMPLES

IN any survey of the different forms of modern power generators it would be unthinkable not to consider the steam locomotive. The steam locomotive is without a doubt one of the most remarkable forms of power generator that has ever been devised. Not only is it entirely self-contained, combining within itself boiler and engine together with ample storage for fuel and water, but while the dimensions to which it can be constructed are strictly limited by the inflexible " loading gauge," which governs both the height and the width of the locomotive, it continues to grow both in power and efficiency, effectively incorporating almost every improvement which is made in the art of engine and boiler construction. For many years past with the appearance of every locomotive of note, engineers have felt that the limit of size and power has surely been reached, but subsequent designs show that still greater powers can be attained. Within recent years, however, the undoubted advance that has been made in the design and construction of locomotives has not been obtained at the expense of any notable increase in size. The height to which the locomotive can be built is limited by the height of the bridges and tunnels under which the locomotive will run, and similarly the width is also fixed absolutely by the width of bridges and tunnels and also by the position of platforms, etc., in the stations. The width of the track is likewise fixed, since old and new rolling stock has to use the same railway lines. These limiting dimensions constitute what is termed the loading gauge of the railway. Although these differ slightly for the various companies operating in this country, the average figures are:

Width of track (measured inside the rails) . . .	4 ft.	8½ in.
Maximum height above rails	13 ft.	6 in.
Maximum width	9 ft.	0 in.

These dimensions, which are shown diagrammatically in Fig. 134, are, it may be noted, practically the standard for the entire world, particularly as regards the width of the track, or the gauge as it is often termed, which is almost universally 4 ft. 8½ in. This is a remarkable tribute to the pioneering work of British engineers in the early days of the development of the steam locomotive, since, both as a result of the large number of foreign railways which were constructed by British engineers and the remarkable influence of British practice on engineering work all over the world, the dimensions originally adopted in this country have, with but few exceptions, been adopted throughout the world. In certain of the colonies a smaller gauge has been adopted, viz. 3 ft. 6 in., because it was originally felt that the traffic to be handled would not demand the employment of the standard gauge. But it is nearly always found that trade follows a railway line, and with the developments which the presence of a railway renders possible the time soon arrives when the desire for a wider gauge is felt. Thus, in Australia, at the time of writing (1924) the question of altering the gauge of some of the railways from 3 ft. 6 in. to 4 ft. 8½ in.

is being discussed, particularly in view of the fact that in certain of the States of the Commonwealth the smaller gauge is employed, and it is desired to develop a trans-continental line in which all trouble due to changing commodities from one set of rolling stock to another will be avoided. On some of the railways in India a 5 ft. 6 in. gauge has been adopted, and provided this could be made the standard for the entire peninsula there would be no reason to regret the adoption of this figure. In the United States of America, the 4 ft. 8½ in. gauge is adopted, but the other dimensions, namely the height and the width, are far more liberal than they are in this country, and this accounts to a very large extent for the fact that American locomotives appear to be much larger than those running on British lines.

Another factor tending to restrict the increase in the dimensions of locomotives on British railways must also be noted. On every line there are large numbers of bridges, viaducts and other structures over which the locomotives run. Many of these were constructed in days when the size of the locomotives, their speed, and the weight of the trains hauled by them were much less than they are to-day. It is obvious that any substantial increase in the loads coming on these structures must tend to induce stresses which are greater than the various parts of the structure can bear. It is not so much the distribution of load over the entire span of a bridge—since many bridges are much shorter than the length of a train—as the extent of the load coming upon any one part, and since the weight of the locomotive is carried by its axles and thence is transmitted to the bridge by the wheels, it is the load on any wheel which must be limited. Hence, as a guarantee of the safety of the bridges as well as of the track itself, we have the rule that the load upon any axle must not exceed 20 tons, that is, 10 tons on each wheel, and this must be strictly adhered to in the design of any locomotive The limitation is applied to locomotives since there is little or no possibility of the weight of carriages or even goods wagons being such as to cause a load of 20 tons on any axle. Thus, limited as regards height and width, limited also as regards the weight that can be borne by any pair of wheels, the locomotive designer has perforce to develop his engines in the only direction left open to him, namely, the length, and it will be found that the tendency of modern design is always towards a greater length of locomotive, thus enabling boilers of greater capacity to be carried, which in turn means that an engine of greater power requiring more steam can be fitted.

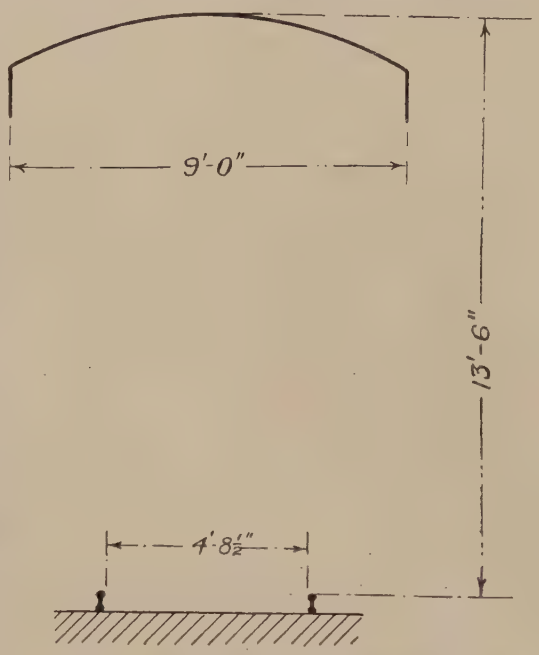

FIG. 134.—Diagram showing the average Loading Gauge for British Railways.

Before we proceed to consider one or two modern examples of railway locomotives, it will be advantageous if we pause for a few moments to study the way in which a locomotive " works." It is undoubtedly a fact that the designers of British locomotives have always paid great attention to the question of appearance, and as far as possible every working part is placed in an unobtrusive position. The result is certainly a very pleasing and well-balanced-looking locomotive, but to a large extent this

accounts for the fact that many people, some of them having a considerable knowledge of engines in general, have little or no idea of the principal features in the design of a locomotive. On the Continent, and to a certain extent in America, locomotive designers are not so reticent, and as a consequence nearly every part of the machinery is exposed to view. From the engineering student's point of view this may perhaps be all to the good, but the result is that in general smartness of appearance foreign locomotives cannot be compared with those running on British railways. Even in the most recent designs, in which the designer, owing to lack of space elsewhere, has been compelled to place certain parts in an exposed position, the same pleasing effect is always aimed at, and, it must be admitted, is generally achieved. The British railway locomotive is still at one and the same time the most efficient, the smartest, and the most pleasing locomotive in the world. It may not be quite in place to speak of beauty in connection with machinery, but it is often possible to produce very pleasing effects in the design of various kinds of engines, and certainly the association of great power with symmetry of outline and smartness of appearance combine in the British locomotive to produce a machine to which the term "beautiful" can be rightly applied.

In order to understand the operation of the steam locomotive, we must remember that it comprises a boiler supplying steam to a reciprocating engine, the engine consisting generally of two cylinders—although three and sometimes four cylinders are employed—working on the non-condensing principle and exhausting the steam to the atmosphere as soon as it has done its work in the cylinders. In some designs the principle of compounding is adopted—that is, the steam after expanding and doing work in one cylinder passes to a second and larger cylinder where it expands further, doing more work; but with very few exceptions compounding is not employed on locomotives running on British railways. Hence we will consider the locomotive in which compounding is not adopted, or to give it its proper technical name, the simple locomotive. The general arrangement adopted in the design of such a locomotive is shown diagrammatically in Fig. 135. We are indebted to Messrs. The Marine and Locomotive Superheaters, Ltd., for this splendid diagrammatic representation of a locomotive, the picture having been specially prepared to demonstrate the arrangement of their superheater as fitted to a modern locomotive. The illustration is, however, admirably adapted for demonstrating the principles underlying the operation of the locomotive, and we will use it for this purpose now, leaving the question of the superheater in abeyance for the moment.

Steam is generated in the boiler as a result of the fuel burned in the fire box, the end of which is on the extreme right of the boiler but not shown in the illustration, the hot gases passing through the tubes into the smoke box and thus away through the chimney into the atmosphere. In order to prevent as far as possible any of the heat generated by the combustion of the fuel from being wasted, as many tubes as possible are fitted between the top of the fire box and the front of the boiler, so as to present the maximum heating surface to the water. These tubes are only fitted in the lower portion of the boiler barrel, since the water only about three-quarters fills the boiler, the remaining space when the boiler is working being filled with steam. In order to prevent the flames from the fire cutting across the lower corners of the tubes, a brick arch is generally erected in the fire box which serves to deflect the flames in such a way that they enter the tubes in a direction parallel to the axes of the tubes.[1]

Owing, however, to the fact that there is a considerable amount of vibration as a locomotive travels along, particularly if the speed is high, it is recognised that there will generally be a large amount of water splashing about in the steam space, and in order that the steam drawn off from the boiler shall be as free as possible from water, it is usual

[1] The reader is referred to Volume IV for a fuller discussion of the subject of boilers. A detailed description of a locomotive boiler will be found in the first chapter of that section of our work.

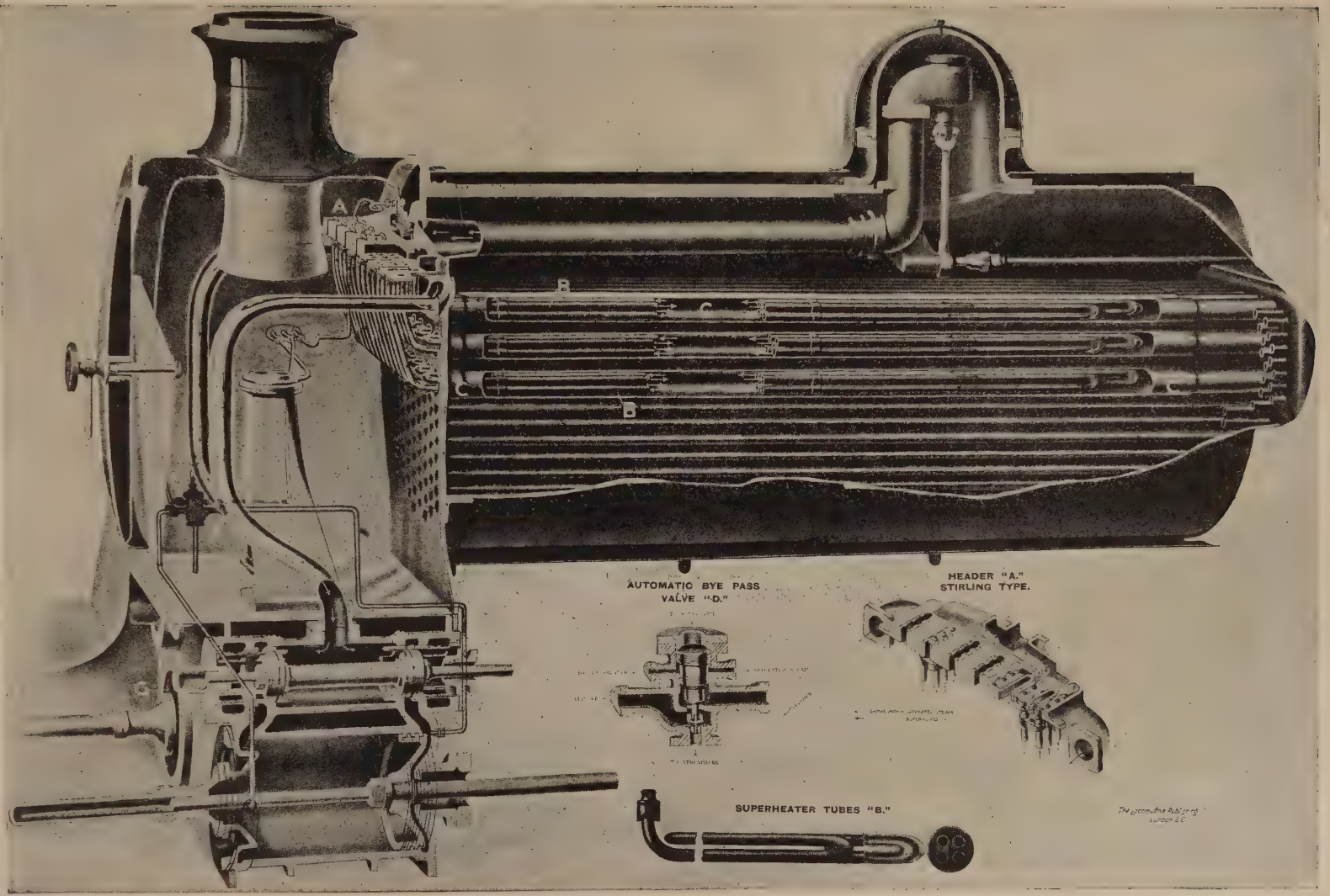

FIG. 135.—Diagrammatic Arrangement of a Modern Steam Locomotive, with Boiler and Cylinder in Section, showing Superheater, Steam Regulator, etc.

to fit a " dome " at some intermediate point along the boiler shell, the interior of which, being further removed from the free surface of the water, forms a more suitable position from which the steam can be withdrawn from the boiler. Moreover, the inside of the dome serves to house the main stop valve, or the regulator as locomotive engineers call it, which controls the supply of steam to the cylinders. The dome is not, however, invariably fitted, certain locomotive designers preferring to withdraw the steam from a point immediately over the fire box, the end of this steam pipe carrying the regulator valve. Thus nearly all the express locomotives of the Great Western Railway are without domes. The steam after it has passed the regulator is led along a pipe fixed inside the boiler, this pipe passing through the front tube plate and out into the smoke box. Attention has already been drawn to the advantages which follow the use of superheated steam in an engine, and, as might be expected, on many modern locomotives superheaters are installed. These generally take the form of a number of smaller tubes which are fitted inside the fire tubes, these fire tubes being of larger diameter in order to accommodate the superheater elements. The general arrangement of the type of superheater designed and constructed by Marine and Locomotive Superheaters, Ltd., is clearly shown in the illustration. The superheater consists of a header, A, which is connected to the regulator pipe, the header being so constructed that it directs the steam through properly arranged channels into the superheater tubes, B, after which it is returned to the header in the direction of the arrows, whence it flows down through the steam pipe, F, into the cylinders, G. An automatic by-pass valve, D, admits a small quantity of steam to the superheater tubes when the engine is stationary and thus prevents the possibility of the tubes being burnt out. The number of the superheater tubes and their length along the large fire tubes depends upon the degree of superheat which it is desired to obtain. If no superheater is installed, the steam passes directly from the front tube plate connection to the cylinders.

It is of interest to note that while nearly all engineers who have to deal with the design and operation of stationary engines have become convinced of the advantages of using superheated steam, there are still many locomotive engineers who prefer to use wet steam. The reason for this is not so much that they are not convinced that superheated steam is far more economical than wet steam from the point of view of steam consumption per horse power, and therefore of coal consumption, but that the difficulties introduced as a result of the higher working temperature in the cylinder more than balance the advantage of greater economy. Where superheated steam is employed, the problem of lubricating the slide valve rubbing surfaces, the cylinder walls, and the stuffing boxes, and the problem of keeping the stuffing boxes steam-tight, become of increasing difficulty according to the degree of superheat employed, and since reliability rather than cost of running is the guiding factor in the design and operation of locomotives, the more reliable wet steam engine is sometimes preferred to the more economical superheated steam engine. But with the progress that has been made in the design of piston rod and valve rod packing for engines using superheated steam, coupled with the use of mechanical lubrication, the disadvantages of superheated steam are being gradually overcome, and it will be found that in the majority of large locomotives now being constructed superheated steam is employed.

The cylinders are mounted on the main frames of the locomotive immediately below the smoke box, and it is in the disposition of the cylinders that the greatest degree of difference between the various types of locomotive will be found. It should be explained that the frames of a locomotive are the two vertical plates, running the entire length of the locomotive, in which the bearings of the wheels are fitted. In the majority of modern locomotives these are to be found inside the wheels, so that as the distance between the flanges of the wheels must be 4 ft. $8\frac{1}{2}$ in., the distance between the frames will be corre-

spondingly less, and where large cylinders are required to develop the requisite amount of power, it is a matter of considerable difficulty to house two cylinders with the necessary valve chests in the small space available. It has already been noted that the locomotive always has two or more cylinders, the reason being that since it must start under load from any position, one cylinder would not be sufficient, as the locomotive will frequently stop with the crank at one or other of the dead centres, in which position it would be impossible to start off again without some external assistance. With two cylinders, however, owing to the fact that the cranks are arranged at right angles to each other, one crank will always be in such a position that steam can be admitted into the cylinder, and thus the locomotive will start off in any position, even if it is necessary to move a short distance in the backward direction first. With the power which is demanded of modern locomotives in view of the weight of the trains hauled and the speed at which they are run, it will be found that with two cylinders only, the diameter of the cylinders will seldom be less than 18 in., a very frequent size being 19 in. diameter by 26 in. stroke. A very simple calculation will show that with two cylinders each 19 in. diameter between the frames there is but little room for anything else.

It is not, however, a question of placing only two cylinders between the frames; we have to remember that valves are necessary to ensure the proper distribution of the steam, and that these valves will work in valve chests which are totally distinct from the cylinders themselves. A glance at the drawings of any of the reciprocating steam engines already described in this volume[1] will show that in many cases the valve chest occupies almost as much space as the cylinder itself, particularly in engines where a single piston valve is employed for dealing with both the steam and the exhaust. In locomotive practice, owing to the necessity for keeping the valve-operating gear as simple as possible, the idea of using separate steam and exhaust valves is never adopted, but a single valve either of the flat D type or of the piston type is employed, a separate valve, of course, being required to each cylinder. In many of the older locomotives when the cylinder diameters were not very great it was usual to employ two flat slide valves working back to back in a common steam chest, the centres of the piston rods and of the valve rods being all in the same plane, the arrangement being shown diagrammatically in Fig. 136. But as the cylinders were made larger, the space available between the cylinders became smaller, and it was almost impossible to place the valves between the cylinders. To surmount this difficulty, the valves were placed either on top of the cylinders, as shown in Fig. 135, or beneath them, and as it was no longer possible to drive them directly by the eccentrics as hitherto, rocking levers were employed. The direct drive of the valve can only take place in a plane containing the crank shaft and the centre lines of the valves, and hence when the valves are placed either above or below the cylinders the valve gear actuates two rods in the plane of the centre of the cylinders, but instead of these rods driving the valves directly, they gave a motion to one end of a rocking lever, the valve being connected to the other end through a flexible joint. Since, however, the motion of one end of the rocking lever is exactly the reverse to that of the other, this method of driving the valves demands a different setting of the eccentrics on the main crank shaft. A great advantage of this arrangement is that it gives plenty of space

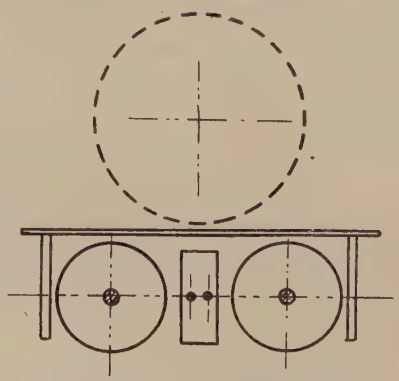

FIG. 136.—Diagram showing the Arrangement of Cylinders and Valve Chest between the Frames of a Locomotive.

[1] The drawing reproduced in Fig. 13 (page 20) may be taken as well illustrating our point.

for the employment of a slide valve having ample dimensions to ensure a proper distribution of the steam without "wire drawing"—that is, the diminution of pressure caused by the steam not having sufficient area to pass through. With the appearance of the piston valve, locomotive designers were not slow to recognise its many advantages over the flat slide valve; and as there was ample space available for two piston valves either above or below the cylinders, it soon became the usual practice to employ piston valves on locomotives.

There was, however, a further difficulty to be faced, even after the problem of arranging for cylinders and valve chests between the frames had been successfully overcome, and this appeared at the main driving axle, i.e. that axle which becomes the crank shaft of the engine. In by far the larger number of locomotives employed in this country, the Stephenson link motion is employed for operating the valve. The outline of Stephenson's valve motion is shown in Fig. 137, from which it will be noticed that there are two eccentrics, these being shown in the diagram by means of their equivalent cranks, OB and OC,

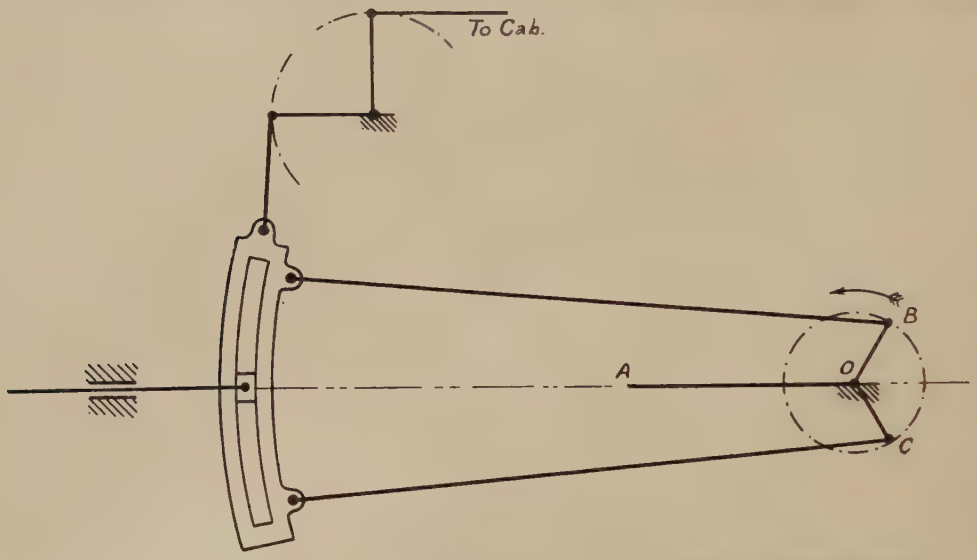

Fig. 137.—Outline Diagram of Stephenson's Valve Motion as fitted to Inside Cylinder Locomotives.

where OA represents the main engine crank. The rods from the eccentrics are connected to the ends of a link as shown, while a block in this link is connected to the valve rod. The link can be moved up or down by a simple mechanism operated from the driver's cab, so that when the link is in its lowest position the block is at the top of the link, opposite one eccentric rod, and thus the eccentric OB directly operates the valve. Similarly, when the link is in its highest position, the eccentric OC operates the valve, and as one eccentric is set for driving the engine in a forward direction and the other for driving it in a backward direction, it is thus possible easily to reverse the direction of running. Moreover, when the link is in an intermediate position, the motion of the valve is controlled partly by one eccentric and partly by the other, and this has the effect of giving an earlier cut-off of the steam to the cylinder. With the valve in full gear, i.e. with the block at one or other extremity of the link, the cut-off of the steam to the cylinder occurs at about 75 per cent. of the stroke, this figure varying with different designs of locomotives. With the block in an intermediate position the cut-off becomes earlier and until when it is at middle of the link there is practically no distribution of steam to the cylinder at all. The earliest cut-off it is possible to obtain is about 10 per cent. of the stroke, and this

occurs when the block is not quite at the mid point of the link part. Beyond mid gear, the cut-off of the steam becomes later and later, but now it is such that it drives the engine in the reverse direction. Thus the valve gear performs the double function of reversing the engine and of enabling a variable cut-off to be employed. Thus when starting the train or when ascending a steep bank the maximum power can be developed, while at other times when the conditions are not so severe the power developed is reduced, thus enabling the engine to be run as economically as possible.

Returning now to the question of the design of the crank shaft, we see that in a two-cylinder engine using Stephenson's valve gear it is necessary to employ four eccentrics, all of which have to be mounted on the main crank shaft between the two cranks. We thus have in the limited space available between the frames two cranks each consisting of two webs with the crank pin between, and four eccentrics, the arrangement being as shown in Fig. 138. When the powers required were not very great, the dimensions of the various parts of the crank and the width of the eccentrics enabled this arrangement to be satisfactorily followed; but with the demand for larger powers, in addition to requiring larger cylinders, larger scantlings of all parts of the driving mechanism were also necessary and it became increasingly difficult to employ the same arrangement as hitherto. As an alternative, the use of a different valve-operating gear was proposed, in which the eccentrics were eliminated. This valve gear, which is known as Joy's valve gear, did not, however, meet with any wide degree of acceptance except on one line, viz. the London and North-Western, for which a number of locomotives were built having inside cylinders and the valves operated by Joy's valve gear.

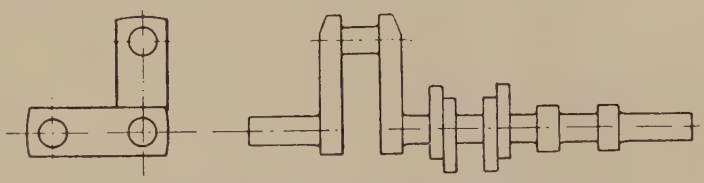

FIG. 138.—Side and Front Elevation of the Crank Shaft of a Locomotive fitted with Stephenson's Valve Gear.

The general outline of Joy's valve gear has already been given (see Fig. 72, page 70), from which it will be noticed that the valve gear receives its motion from a point on the connecting rod, whence, operating through a series of rods, a much smaller motion is given to a block working on a link, which in turn gives its motion to the valve. The functions of this link are exactly the same as those of the link in the Stephenson gear, viz. to reverse the engine and also to provide for a variable cut-off of the steam to the cylinder; but in this case these functions are performed by rotating the link about an axis as shown in the diagram, and not by raising or lowering it. In the position shown by means of the full lines the link is in full backward gear, i.e. it will control the motion of the valve so that the engine runs in the backward direction, the cut-off occurring as late as possible, again about 75 per cent. When the link is in the position shown dotted, the engine moves in the forward direction at maximum cut-off, while with the link practically vertical, i.e. in mid gear, there is no distribution of the steam. Intermediate positions between full gear and mid gear give an earlier cut-off of the steam. It will be noticed that the valve is driven directly in a position above the cylinders. Using Joy's valve gear, the cranks can be made of substantial dimensions and cylinders of the largest possible diameter that can be fitted between the frame plates can be used. But with the exception noted above, locomotive designers have not taken kindly to Joy's valve gear, although experiments with this gear have been made on a number of lines. Thus, faced with the difficulty of fitting cylinders of sufficiently large diameter between the frames, and of accommodating the large cranks and eccentrics required along the main driving axle, many designers adopted the expedient of placing the cylinders outside the frames, and thus we obtain the familiar outside cylinder locomotive.

In principle, the outside cylinder locomotive does not differ in any way from that in which the cylinders are inside, but in adopting the former expedient the designer has at once placed at his disposal a far greater amount of space than he had when he crowded everything between the frames. There are thus many different arrangements adopted in the design of outside cylinder locomotives. The cylinders are bolted outside the frames, and in many cases the valve chests are inside the frames, while in others both the cylinders and the valve chests are outside the frames. The valve gear itself—and now with plenty of space available this is generally of the Stephenson type—is placed inside the frames, the driving axle carrying only the four eccentrics, two adjacent to each frame. A direct drive to the valve can be provided when the valve chest is inside; but with outside valves, a rocking lever is provided which transmits the motion through the frame and thence to the valve rod. Special provision must be made for supporting the crosshead guide bars, a bracket bolted to the frame and to the underside of the footplate being

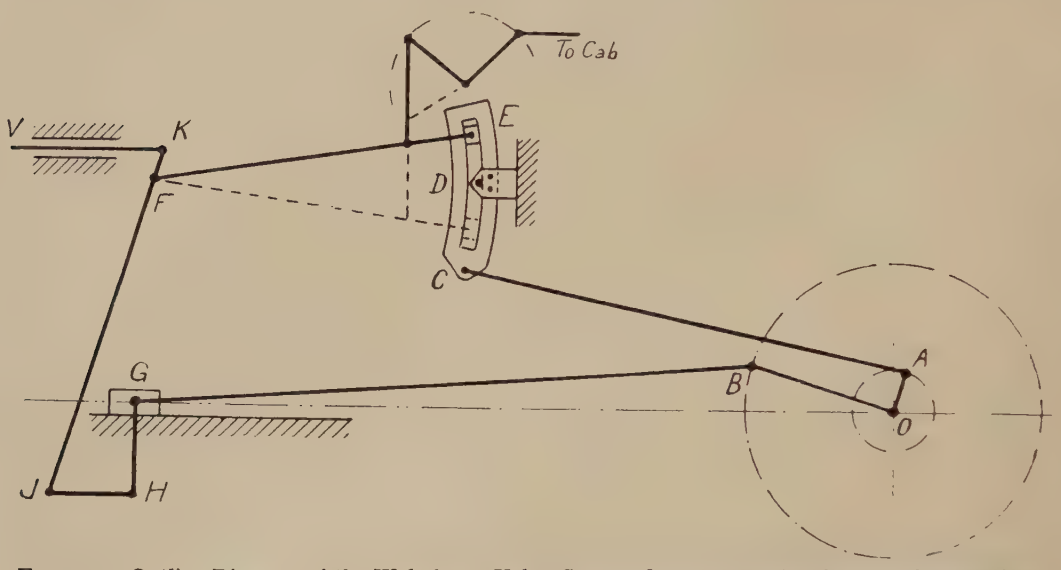

Fig. 139.—Outline Diagram of the Walscheart Valve Gear as fitted to Outside Cylinder Locomotives.

generally employed at the outer end, the inner ends being secured to the cylinder cover.

With the adoption of the outside cylinder principle, the possibility of employing more than two cylinders became at once apparent. For many years locomotive designers did not find it necessary to employ a greater number of cylinders than two, and it was only within recent years that the demand for greater power led to the adoption of three or even four cylinders. In these latter designs, the extra cylinders are placed inside the frames, so that the arrangement becomes not dissimilar to a combined inside and outside cylinder arrangement. In three-cylinder locomotives the cranks are arranged at 120° to each other and three separate valve motions must be provided. Little or no difficulty is experienced in fitting all the gear and the one central crank between the frames. With the four-cylinder engine, in which the cranks are all at 90° to each other, a certain amount of trouble would be experienced if four distinct sets of valve gear had to be fitted, which, if the Stephenson gear is adopted, would necessitate the use of eight eccentrics. To get over this difficulty, only two sets of gear were provided, two valves being driven from one motion, the first directly and the second through a rocking lever. With two of the cranks at 180° to each other the pistons are moving in exact opposition to each other, and

therefore the valve driven through the rocking lever will also be in exact opposition to that driven direct from the eccentrics.

The need for a valve gear which could be placed outside the cylinder with the crank and connecting rod of the outside cylinder was, however, severely felt, and many attempts were made to surmount the obvious difficulties in the way of providing such a gear. The writer recalls seeing on the Continent many years ago an outside cylinder locomotive in which the two eccentrics, link, and reversing gear, all complete, were outside the frames, exposed "naked and unashamed" to view. This, as already indicated, is more or less in accordance with foreign ideas in the design of locomotives, but to British engineers the arrangement appears to be so very ugly that it would hardly be tolerated in this country. The Joy's valve gear has been employed on outside cylinders with more or less satisfactory results, but while, as far as appearance goes, this arrangement leaves but little to be desired it has not proved entirely satisfactory. A new form of valve gear was, however, introduced on outside cylinder engines, which, while at first it appeared rather foreign to our ideas of what was fit and proper in locomotive design, has gradually found considerable favour among locomotive designers on British railways. This is known as the Walscheart valve gear—a name which indicates its Continental origin. The outline of this gear is shown in Fig. 139, from which it will be noticed that the motion is obtained by a combination of the motion of a single eccentric and of a point driven from the cross-head. The point A describes a circle of small radius around the crank-shaft centre, OA being at right angles to the main crank OB, so that the motion of the rod AC is the same as if it were driven by an eccentric set at 90° to the main crank. AC gives a rocking motion to the reversing link about its centre D. The block E working in this link can be moved up or down in the link by an arrangement of links as shown operated from the cab of the locomotive, thus reversing the engine or varying the point of cut-off of the steam to the cylinder. The valve is not, however, driven directly from this block, the other end of the rod, EF, receiving a motion from the crosshead, G, through the rigid bracket, GH, operating through the links HJ and JK, KV being the valve rod. This mechanism, while at first sight it appears to be rather complicated, is indeed particularly simple, and apart from the fact that it can be fitted entirely outside the frames and that it gives a very good distribution of the steam, it possesses the advantage that it is far lighter than any of the alternative forms of gear. On express engines this is of considerable value, as the wear-and-tear of the bearings on heavy links moving with reciprocating motion at high velocities is very severe. It is possibly this reason more than any other which accounts for the popularity of the Walscheart gear at the present time.

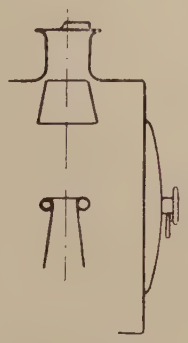

FIG. 140. — Diagrammatic Arrangement of the Smoke Box of a Locomotive, showing position of Blast Pipe.

In every type of steam locomotive, no matter what the arrangement of cylinders may be, the steam after it has done its work in the cylinders is led to a single pipe which is placed in the smoke box immediately beneath the chimney with its centre line vertical and coincident with the centre line of the chimney. This pipe is known as the blast pipe, and the steam issuing from it, particularly when the cut-off in the cylinders is late, passes through the chimney with considerable pressure and speed, giving rise to the familiar "puff" of the engine. When the cut-off occurs early in the stroke, as the steam has expanded to a considerable extent in the cylinders, the pressure at which it is exhausted to the blast pipe is much lower, and both the sound and the effect of the puff are considerably reduced. To the inside of the chimney is fitted a conical pipe which extends downwards a considerable distance into the smoke box towards the top of the blast pipe,

as shown in Fig. 140. Thus, the jet of steam issuing from the blast pipe is caught in the conical portion of the chimney, where it has a powerful ejector effect, drawing a large volume of the hot gases from the tubes with it. This tends to create a vacuum in the fire box, and a current of pure air rushes into the fire box either through the fire box door and over the burning fuel, or under the fire box through the fire bars and thus through the incandescent fuel. In this way a very effective draught of air is created with every puff of steam from the chimney, and the rate of combustion of fuel in the fire box is considerably increased.

The reader has probably noticed on a dark night particularly on slow-moving goods engines that, when the stoker has the fire box door open, the reflection of the fire on the trail of steam above the engine is momentarily brighter with each puff from the chimney. This simple observation serves to prove the value of the blast in assisting rapid combustion of the fuel and therefore rapid production of steam. The importance of the correct design and setting of the blast pipe cannot be over-emphasised, as without it, unless of course some alternative system were adopted, there would be no possibility of obtaining a boiler in the space available which could develop steam sufficiently rapidly to meet the requirements of a high-powered modern express engine. It is worthy of notice that George Stephenson adopted the idea of the blast pipe on his celebrated engine *Rocket*, which won the historical Rainhill trials in 1829, fitting a conical nozzle to the end of the exhaust pipe where it terminated inside the chimney. The remarkable success of the *Rocket* in these trials has been thought to be due in no small degree to the remarkable effect of this simple device, coupled with the adoption of a multitubular boiler very similar in form (though very much smaller) to that employed to-day. Certainly the *Rocket* laid the foundation stone of modern locomotive practice. But to revert to modern conditions. When the locomotive is at rest, the blast due to the exhaust from the cylinders is necessarily absent, and as it is frequently necessary to have a draught through the furnace, such as when preparing for the start of a long journey or when raising steam from cold, a ring is generally formed round the top of the blast pipe to which is connected a small pipe conveying steam direct from the boiler. This ring is shown in Fig. 140, and the reader can also turn back to Fig. 135 where the small pipe conveying steam to the ring is clearly shown. A number of holes are provided on the top surface of this ring, so that the driver can, at will, send a number of small jets of steam up the chimney. These have the effect of drawing air through the fire box and thus maintain a small but necessary draught of air over and through the burning fuel. The familiar "purring" of a locomotive in the station which is generally accompanied by the discharge of a small quantity of steam from the chimney is due to the use of the auxiliary blast.

In the three- and four-cylinder locomotives referred to above all the cylinders receive steam direct from the boiler and exhaust it direct to the atmosphere. We have already seen in discussing the development of the stationary engine that, as a means of obtaining more economical results, compounding is nearly always adopted, and that so great are the advantages which follow the adoption of the principle of compounding that we frequently have not merely two successive expansions of the same steam, but three and even four, as in the triple-expansion and the quadruple-expansion engines. The question at once arises, is compounding adopted in locomotive practice? For a number of years various attempts have been made by locomotive designers to introduce the compound locomotive on British railways, but all these attempts have apparently ended in failure. The reason for this is all the more difficult to understand when it is remembered that in the United States of America and on the continent of Europe compounding is very extensively employed on high-powered locomotives. It cannot be said that British locomotive engineers have lightly brushed aside the idea of compounding; it has been

carefully tried on a number of railways and only after a thorough and complete test has it been discarded in favour of the " simple," i.e. non-compound engine.

The most notable attempt to employ compounding on locomotives was that made by the late Mr. Webb on the London and North-Western Railway. During the period of his tenure of the office of locomotive superintendent of this line, a large number of compound engines were built, having three cylinders, viz. two high-pressure cylinders outside the frames which exhausted into one large low-pressure cylinder placed inside the frame. On others four cylinders were employed, two high-pressure and two low-pressure cylinders. All these locomotives worked well for a time, and of course their economy from the point of view of coal consumption was very good indeed; but with the larger and heavier trains that were required to fulfil the traffic requirements of the North-Western line it became necessary to employ two engines on nearly all the express trains, so that the economy disappeared. As a result, with the appearance of a successor to Mr. Webb, the compound engines were gradually withdrawn and their place taken by larger and more powerful simple locomotives. Again, on the Midland line compounding was carefully tried, a number of alternative designs being built, but here also the idea was not a complete success. On the North-Eastern Railway a number of compound locomotives were built, but after a thorough trial they were discarded in favour of the non-compound type. But the most conclusive experiment was that made by the locomotive superintendent of the Great Western Railway. Recognising the fact that compound locomotives were being run successfully on the Continent, he purchased a four-cylinder compound locomotive from a famous firm of French locomotive engineers which was practically identical with those which were actually running on some of the railways in France, and at the same time he built a simple engine of approximately the same dimensions. The performances of these two were carefully studied under the actual running conditions as found on the G.W.R. Two additional compound locomotives were subsequently bought from France, but although the results obtained from these French compound locomotives were not unsatisfactory, the conclusion was ultimately arrived at that they were not suited for the service on British lines, and all subsequent locomotives built for the Great Western line have been non-compound, many of them with four cylinders. These engines are among the fastest and most powerful in the country, this line having the honour of operating the longest regular non-stop run in the world, viz. from London to Plymouth, a distance of 226½ miles in 4 hours 7 minutes—that is, at an average speed of just over 55 miles per hour.

A number of reasons have been advanced from time to time to explain why it is that the compound locomotive is not suited for use on British railway lines. One of the most convincing of these is that, owing to the limitations imposed by the loading gauge, the power developed by the compound engine is less than that of the corresponding simple engine, and thus the locomotive is not able to deal with the loads which constitute the ordinary everyday trains in this country. Thus, if we had a simple locomotive with two cylinders 19 in. in diameter with a piston stroke of 26 in.—not an exceptionally powerful locomotive this—by adopting the compound principle, in order to obtain the same power we should require two high-pressure cylinders, of, say, 16 in. diameter, and these would exhaust into two low-pressure cylinders of about 21 in. diameter, the stroke in each case being the same, viz. 26 in. The reader will at once recognise that to fit these four cylinders in the small width available will be a difficult problem indeed. Attempts have been made to overcome this difficulty by having the cylinders at a different level, the two high-pressure cylinders raised above the two low-pressure cylinders, but the arrangement tends to become unduly complicated. Again, the gradients on British railway lines are notoriously severe, and moreover they change frequently on any given journey. This demands a very flexible locomotive, i.e. one that can now develop great powers

and a few minutes later develop only a small power. The compound locomotive, in view of its smaller high-pressure cylinder area, cannot be so flexible as the simple engine with a considerably larger high-pressure cylinder area, since in this case the entire cylinder area may be considered as constituting high-pressure cylinder area. Of course the disadvantage which was advanced a few years ago, that with its three or four cylinders the compound locomotive becomes very complicated and that with the multiplication of connecting rods, valve rods, etc., the wear-and-tear is excessive, does not now apply, since the three- or four-cylinder simple locomotive is becoming more and more common on the great trunk lines of this country.

In every type of locomotive, no matter what the size and number of the cylinders may be, there will be found certain details to which attention must now be briefly drawn. In connection with the boiler itself there are the usual fittings which form a part of every boiler. These comprise the safety valve, which is generally mounted at the rear of the boiler over the fire box, the gauge glasses and the pressure gauge, these being placed inside the cab. Provision must, of course, be made for maintaining the height of water in the boiler at a proper level, and for this purpose some form of pump is installed or else what is termed an " injector " is employed, in which by the use of a jet of steam from the boiler itself a stream of water is carried into the boiler. Either feed pumps or injectors are fitted in duplicate on the majority of locomotives, while in others a pump and an injector are fitted. In addition, we find on all locomotives which haul passenger trains an arrangement for providing the pressure which operates the continuous automatic brake on the train. This may take the form of an air pump, as is found on the locomotives of those railways where the Westinghouse brake is employed, or of an ejector producing a vacuum (which may be considered as producing negative pressure) on locomotives where the vacuum brake is employed. In the latter case the pressure which operates the brakes is produced by the atmosphere working against the vacuum. Other fittings and refinements, such as power-operated reversing gear, mechanical lubricators, etc., will be found, but the reader must be referred to textbooks dealing with exclusively locomotive design for a fuller treatment of this subject.

It may be noted in conclusion that because compounding is found economical on large stationary engines such as factory engines and on marine engines, it does not therefore follow that it should prove economical on locomotives. The locomotive differs fundamentally from both of the types mentioned in that it is a non-condensing engine. Attempts have been made to incorporate the condenser in the design of a high-powered locomotive, but with one or two exceptions [1] the idea has not been seriously entertained. Thus it follows that the exhaust pressure, even when a very early cut-off is employed, can never be less than about 3 lb. per sq. in. above atmospheric, i.e. 18 lb. per sq. in. absolute, as compared with, say, 2 lb. per sq. in. absolute (i.e. 26 in. vacuum) in the case of a condensing engine having condensing plant of quite moderate dimensions. Thus the limit of expansion between boiler pressure and exhaust is necessarily reduced in locomotive practice, and this may be held as accounting, at least partially, for the fact that British locomotive designers do not consider that compounding is necessary. Of course, the same figures apply to the American and Continental locomotives, but there the peculiarities of the conditions under which the engines run, the loading gauge, etc., may be held as having considerable influence on the problem. In any case, it cannot be gainsaid that British locomotives are among the fastest and the most reliable in the world, and reliability rather than great economy in fuel consumption must always be the chief aim in locomotive design.

Locomotives are often classified in accordance with the arrangement of wheels adopted

[1] See Volume I, Chapter II, for a description of the Reid-Ramsey turbo-electric locomotive, on which a condenser is fitted.

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PLATE V.—THREE-CYLINDER "PACIFIC" TYPE EXPRESS LOCOMOTIVE FOR THE GREAT NORTHERN SECTION OF THE LONDON AND NORTH-EASTERN RAILWAY.

for transmitting the power developed and for carrying weight of the various parts of the locomotive itself. Although, generally speaking, we have only one crank shaft, even when three or four cylinders are employed, it is found that if the power developed at the crank shaft is transmitted by means of only those wheels which are mounted on the crank shaft, undue slipping is very liable to occur. Particularly on wet and greasy rails, if the power at the cranks is considerable, it is easy to see that with merely the friction at two points of contact the possibility of the wheels gripping the rails is very small. Hence in all modern locomotives it will be found that in addition to the wheels on the crank shaft—or the driving axle, as it is usually termed—we have wheels of the same diameter mounted on the adjacent axles, these wheels being coupled together by means of what are known as coupling rods working on cranks of the same radius as the main driving cranks. Where two pairs of wheels are coupled in this way we have what is termed a four-coupled engine ; with three pairs of wheels coupled, a six-coupled engine, and so on. The more pairs of wheels that are coupled, the greater " adhesive power " does the locomotive possess, i.e. the greater force can it exert to haul the train without slipping. On the other hand, coupling the wheels in this way militates against its capability of running freely at high speeds. The old " single " engines, i.e. those engines that merely had a single pair of driving wheels, were able to run with extreme smoothness at very high speed, but they could only deal with light trains, and thus they gradually became unsuited for the conditions met with to-day on the majority of British lines. They were, however, to be found on nearly all the chief British railways twenty or thirty years ago, and as far as speed was concerned they put up many performances which have hardly been excelled by the giant locomotives which have superseded them. But on all modern express locomotives we have either four- or six-coupled wheels, while for the heavier locomotives which deal with the goods traffic eight- or ten-coupled wheels are often employed.

In addition to the driving wheels, we have to provide some means for supporting the cylinders and the front end of the boiler, and in view of the position of the former, wheels of small diameter must be employed. Moreover in view of the fact that the front wheels are responsible for the negotiation of all curves, a certain degree of flexibility is desirable at this point. Hence we generally find that the front part of the locomotive is carried on what is termed a " bogie," i.e. two pairs of small wheels mounted on a frame which is free to swivel about a pivot at the centre of the frame. Beneath the fire box and the cab at the rear end of the frame, it is often necessary to provide a pair of wheels other than the rearmost pair of driving wheels, and these, too, are of fairly small diameter and are known as the trailing wheels. Thus, in a locomotive having a leading bogie, two pairs of coupled wheels, and a trailing pair of wheels we have an arrangement which forms what is known as the " Atlantic " type engine, the wheel arrangement being written thus : 4–4–2. Other familiar wheel arrangements are not always given special names, but they are always written in the manner shown, in the order, number of leading wheels, number of driving wheels, number of trailing wheels. The wheel arrangements generally employed on locomotives are shown in Fig. 141.

Another problem which faces the locomotive designer, apart altogether from the development of power, is that of finding ways and means for carrying sufficient coal and water for the requirements of his locomotive. When it is remembered that an express locomotive will consume about 1 ton of coal on a journey of 50 miles, the necessity for carrying a considerable weight of coal is at once apparent. Similarly with the continual wastage of water in the form of exhaust steam passing up the funnel, a reserve of water must be carried to meet the continual requirements of the boiler, which can be roughly estimated as 1,500 gallons per ton of coal burned. On large express engines, the coal and the water are usually carried in a separate vehicle known as the tender, which is

attached to the locomotive, the coal being placed above the tank in which the water is stored. The coal- and water-carrying capacity of the tender will depend upon the special circumstances of the line upon which the locomotive runs, such as the length of non-stop journeys. Where long non-stop journeys have to be performed, it is usual to find that water troughs are installed at different points along the line, so that the locomotive as it runs along can pick up a supply of water by the simple expedient of dropping a scoop into the trough. Under these circumstances a larger portion of the available space in the tender can be utilised for the carriage of coal, since no satisfactory method of picking

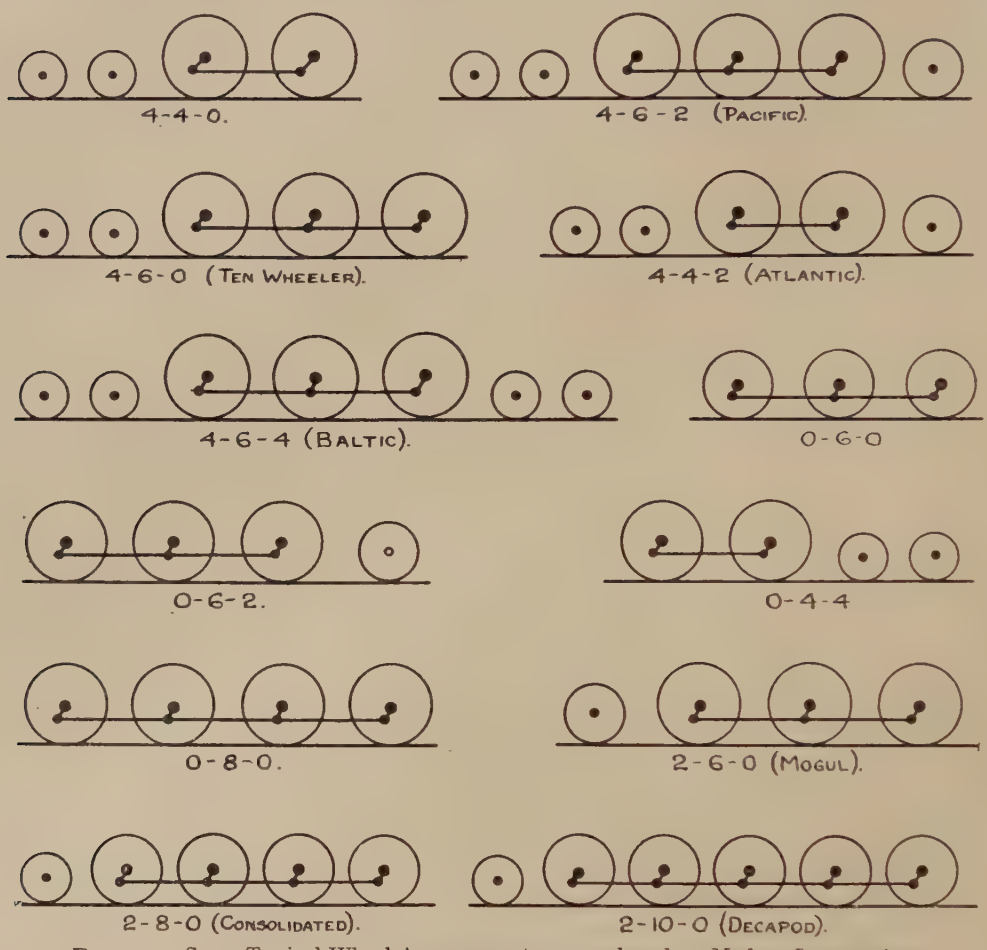


Fig. 141.—Some Typical Wheel Arrangements as employed on Modern Locomotives.

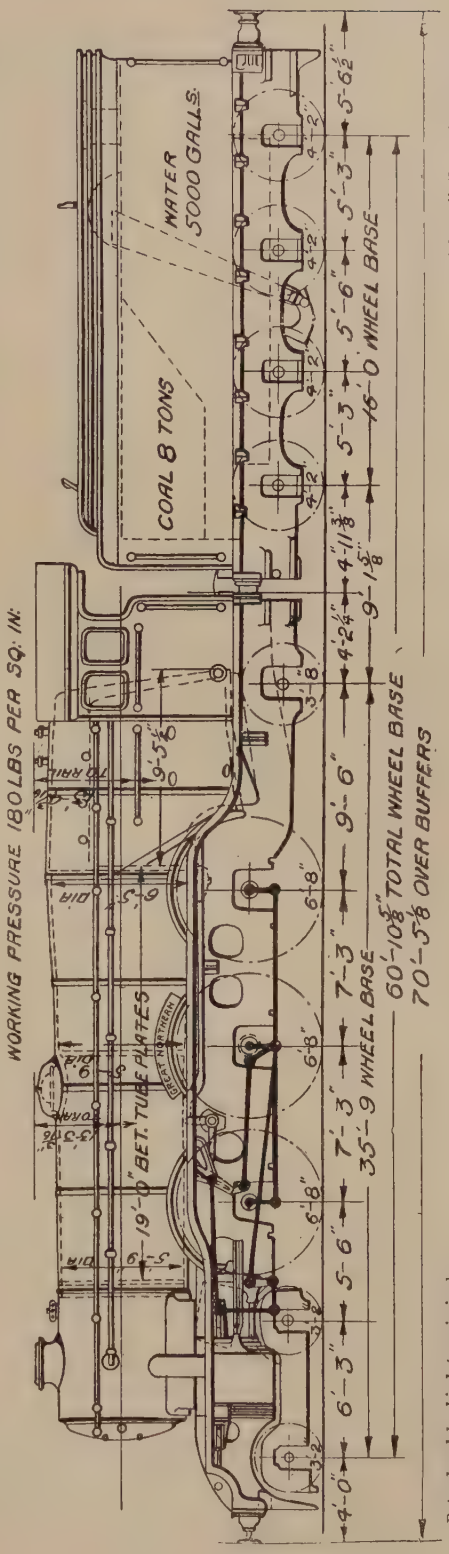
up coal while the locomotive is in motion has yet been devised. Thus in a large tender for a modern locomotive it is possible to carry 6 tons of coal and 4,000 gallons of water, and even these figures are sometimes exceeded.

Where the non-stop journeys that have to be undertaken by the locomotive are not so great, the tender can be dispensed with and the necessary supplies of coal and water carried on the same frame as the engine itself. In this case, the coal is generally stored in a bunker behind the driver's cab and the water is carried in tanks mounted one on each side of the boiler in front of the cab. Sometimes water is also carried in a tank forming the lower part of the bunker, and sometimes a tank is carried at the rear of the engine between the frames. In any case, when the engine is not provided with a tender it is

known as a "tank" engine, and to distinguish between the two, in giving the wheel arrangement it is usual to introduce the letter T thus: 4–6–2T. This would signify a tank engine having a wheel arrangement consisting of a leading bogie, six coupled wheels and a trailing pair of wheels, and constitutes what is termed a "Pacific" type engine. A feature of the tank engine is the enclosed cab, which enables it to be worked equally well in either direction. For short journeys the necessity of turning the engine on a turn-table is thus obviated, although it should be noted that with the regulator, whistle, control, etc., at the front of the cab, it is always easier to drive the engine in the forward direction, i.e. with the chimney leading.

We have endeavoured in the above paragraphs to give a brief description of the principles underlying the design of all types of locomotives and to point out the various reasons which have led to the development of different types. Locomotive engineering forms, however, a distinct branch of engineering, having an extensive literature of its own, and it has not been possible to touch more than the fringe of the problem, and the student who is particularly interested in this phase of our subject will find a number of books available from which he can study the theory and practice of locomotive design at far greater length. Sufficient has, however, been said to enable the reader to understand the fundamental features of the design of a locomotive, and we can now proceed to describe a few examples representing some of the various types of modern locomotive which are being widely used at the present day.

We will consider first of all the locomotive shown on Plate V. This photograph, for which together with the other illustrations of this locomotive we are indebted to the editor of the *Railway Gazette*, shows a three-cylinder express locomotive of the 4–6–2 or "Pacific"



[from "The Railway Gazette."

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FIG. 142.—General Arrangement Drawing of Three-cylinder "Pacific" Type Express Locomotive for the Great Northern Section of the London and North-Eastern Railway.

type, designed for use on the Great Northern section of the London and North-Eastern Railway by Mr. H. N. Gresley, the locomotive engineer of the Great Northern line. The principal dimensions are given in the drawing reproduced in Fig. 142, while the position of the cylinders and the overall dimensions of the engine are shown in the end elevation drawing Fig. 143. As already noted, the locomotive has three cylinders, two of which are outside the frames with their centre lines horizontal and level with the centre of the driving wheels, while the third cylinder is placed between the frames at a slope of one in eight towards the driving axle, so that the connecting rod shall clear the front driving axle, all three cylinders driving on to the middle pair of coupled wheels. The middle cylinder is indicated by dotted lines in the drawings. The engine is not operated on the compound principle, each of the cylinders, which are 20 in. diameter with a piston stroke of 26 in., receiving steam direct from the boiler. The distribution of steam is effected by means of only two sets of valve gear, this being of the Walscheart type operating directly the valves of the outside cylinder, while by a simple arrangement of levers patented by Mr. Gresley, the valve for the middle cylinder, which is placed to the left of the cylinder looking forward, is actuated from the adjacent valve gear. The valves are of the piston type each 8 in. diameter. There are three pairs of driving wheels coupled together, the axle of the second pair forming the main driving axle. The diameter of the driving wheels is 6 ft. 8 in. A bogie having wheels 3 ft. 2 in. diameter carries the front part of the locomotive, while the rear end is carried on a single pair of wheels 3 ft. 8 in. diameter, the axle boxes of which are so arranged that the wheels have a side play of 2½ in. each way. The distances between the various centres of these wheels are given in Fig. 142, from which it will be noticed that the length overall of the locomotive, excluding the tender, is 43 ft. 11¼ in.

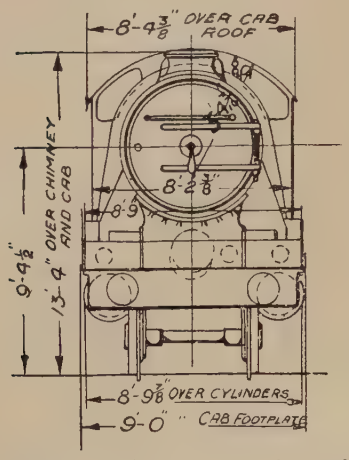

FIG. 143.—Front Elevation of "Pacific" Type Locomotive for the London and North-Eastern Railway, showing Position of Cylinders.

The boiler is built up in three parts, the fire box casing and the barrel, this latter comprising a conical portion adjacent to the fire box, and a cylindrical portion at the front end. A photograph of the boiler as it appeared when completed, but prior to being erected on the locomotive frame, is reproduced in Fig. 144. The length of the boiler overall is 28 ft. 5½ in., while the barrel is 19 ft. long, this being the length between tube plates. The conical portion is 8 ft. 7⅜ in. long and is 6 ft. 5 in. diameter at the tube plate end tapering down to 5 ft. 9 in. diameter, this being the diameter of the cylindrical part, which is 11 ft. 9⅝ in. long. The boiler is fitted with 168 ordinary smoke tubes, having an outside diameter of 2¼ in., and in addition there are 32 tubes each 5¼ in. outside diameter for housing the superheater elements. The fire box, which is of what is sometimes referred to as the Wootten type, is designed to give the maximum grate area with the minimum length. A side view of the fire box before it was placed in the boiler is shown in Fig. 145; while Fig. 146 shows a front view of the fire box. In these views the foundation ring and the holes for the staybolts, by means of which the fire box is connected to the outer shell, can be clearly seen, and attention may also be drawn to the rear tube plate in the latter illustration, in which the relative positions of the superheater tubes and the ordinary tubes are clearly shown. The fire box has a grate area of 32 sq. ft., while the total heating surface of the boiler is 2,930 sq. ft., comprising the following:

Boiler heating surface :—

	Sq. ft.
Fire box . . .	215
Tubes (small) . .	1,880
Tubes (large) . .	835
Total . .	2,930

The total heating surface of the superheater, the elements of which are 18 ft. 5⅝ in. long, is 525 sq. ft., and the working pressure in the boiler is 180 lb. per sq. in.

The driving cab is exceptionally spacious, and so arranged as to afford the greatest protection to the driver and the fireman. As will be seen from Plate V, the sides extend for almost the full length of the foot plate, side windows being provided. The roof extends well back over the tender, thus affording protection to the fireman when he shovels coals from the front of the tender. Great care has been expended on the arrangement of the various regulators and controls in the cab to ensure the maximum efficiency. A view of the inside of the cab is shown in Fig. 147. The main regulator control is duplicated, so that it can be operated from either side of the foot plate by the simple action of pulling up a lever. The various feed regulators, vacuum and hand brake control levers, the whistle, etc., are all placed within easy access and a very clear look-out is provided at the front of the cab. It may also be noted, as indicating the forethought that has been expended in the fitting up of the cab, that padded seats are provided for both the driver and the fireman; so that their long and arduous spells of duty can be performed under the most comfortable conditions possible.

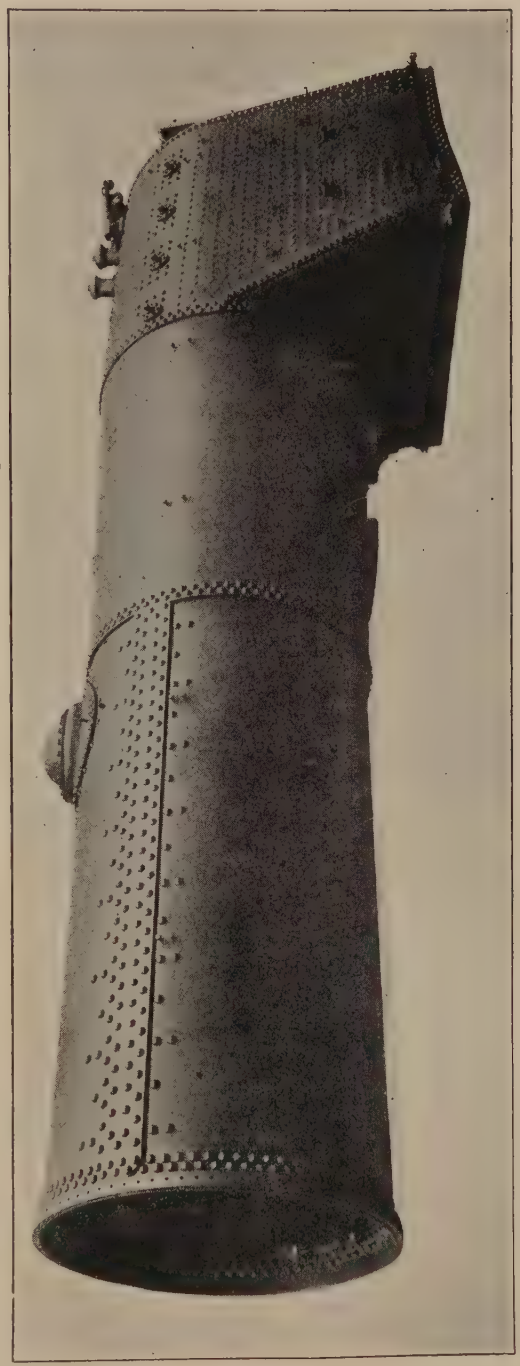

Fig. 144.—Boiler for "Pacific" Type Locomotive for the London and North-Eastern Railway prior to being placed on the Frames.

The tender is designed for a maximum capacity of 5,000 gallons of water and 8 tons of coal. The Great Northern line is fitted with water troughs at various points, and a water pick-up arrangement is therefore fitted to the tender, so that the supply of water can be augmented from time to time. The tender is carried upon four pairs of wheels without bogies, each wheel being 4 ft. 2 in. diameter, and the rigid wheel base 16 ft. The weight of the tender in working order is 56 tons 6 cwt. and the length overall is 26 ft. 5⅞ in.,

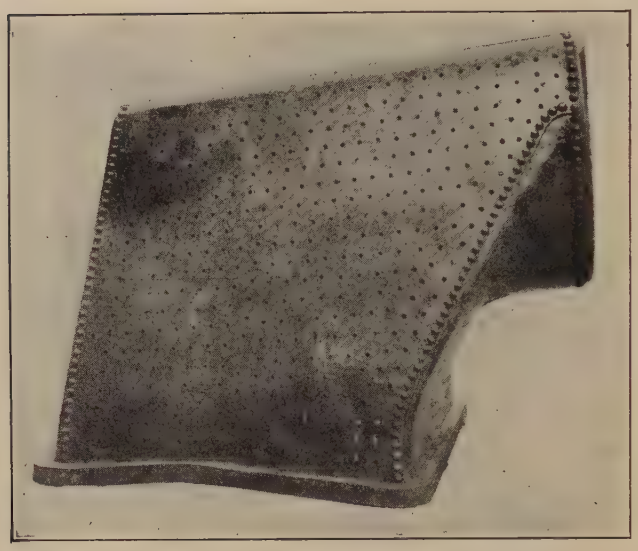

Fig. 145.—Side View of Fire Box for "Pacific" Type Locomotive for the London and North-Eastern Railway.

so that with the length of the engine itself, viz. 43 ft. 11¼ in., we have an overall length for the engine and tender of 70 ft. 5⅛ in. The total weight of the engine is 92 tons 9 cwt., distributed as follows: 17 tons 1 cwt. on the bogie, 20 tons on each pair of coupled wheels, and 15 tons 8 cwt. on the trailing wheels. The weight of the engine and tender is thus 148 tons 15 cwt. The engine bears the name "Great Northern" and is numbered 1,470. She has a very smart appearance, as will be gathered from the photograph reproduced on Plate V, while an impression of her massive strength is obtained from Fig. 148, which is a view looking right on the front of the locomotive. Taken altogether, this may be regarded as a splendid example of the type of express locomotive as used on the premier British lines at the present day.

As an example of the tank engine we may consider briefly one of a class that has been designed by Mr. Billinton for the London, Brighton and South Coast section of the Southern Railway. This engine, a photograph of which is reproduced on Plate VI, is of the 4–6–4T, or "Baltic" type, having a bogie at both ends in addition to the six coupled wheels, these latter being 6 ft. 9 in. in diameter. The principal dimensions and the distribution of the weights are given in the outline drawing reproduced in Fig. 149. The engine has two cylinders, each 22 in. in diameter, with a piston stroke of 28 in., the cylinders being outside the frames with their centre lines 6 ft. 7½ in. apart. The valve gear is of the Walscheart type actuating the valves, which are placed inside the frames, through rocking levers. The centre lines of the valves are 2 ft. 6 in. apart. Superheated steam is supplied at a pressure of 170 lb. per sq. in. from a boiler which is 5 ft. 4¼ in. diameter and 15 ft. 1⅛ in. long between the tube plates. The fire box, which is of the "Belpaire" type, is 8 ft. 6 in. long, the overall length of the

Fig. 146.—Front View of Fire Box showing Tube Plate.

boiler being 23 ft. 8¾ in. There are 173 tubes in the boiler, of which 21 contain the superheater elements, these tubes being 5½ in. in diameter. Of the remainder, 142 are 2 in. diameter and 10 are 2¼ in. diameter. The total heating surface is 1,816·56 sq. ft. obtained as follows:

Boiler heating surface:—

	Sq. ft.
Tubes (small)	1,211·15
Tubes (superheater)	453·33
Fire box	152·08
	1,816·56

The grate area is 26·68 sq. ft. Water is carried in the side tanks and also in the lower portion of the coal bunker, the position of the tanks being indicated by dotted lines, this arrangement constituting what is often termed a "double-ended tank." The two side tanks are connected together by means of a large pipe running beneath the boiler, and they are also connected to the end tank by means of pipes situated just under the foot-plate. One of these pipes is clearly shown in the photograph. The water-carrying capacity is thus very large for an engine of this type, being no less than 2,686 gallons. This large water-carrying capacity is called for on account of the fact that there are no water troughs on the London, Brighton and South Coast lines. In the bunker there is space for the storage of 3½ tons of coal.

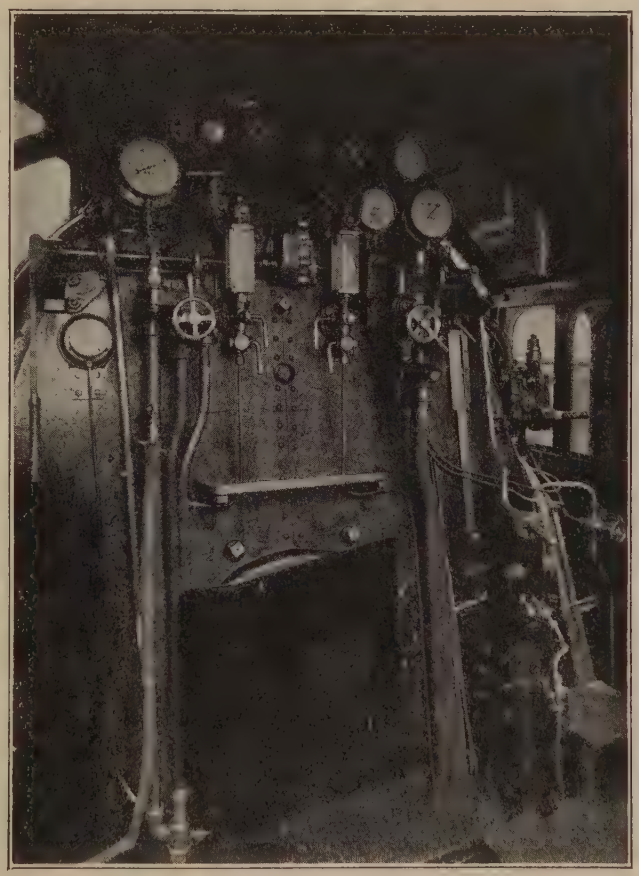

FIG. 147.—The Interior of the Cab of the "Pacific" Type Locomotive, London and North-Eastern Railway.

The distribution of the weight on the various axles is shown in the drawing (Fig. 149), from which it will be noticed that on the centre driving axle, i.e. on the main driving axle, the load is 19 tons 5 cwt., thus nearly reaching the maximum of 20 tons, the weight on the other driving axles being slightly less, the leading bogie also carries 19 tons 5 cwt., i.e. 9 tons 12 cwt. 2 qr. on each axle, while the weight on the rear bogie is 11 tons 5 cwt. per axle. These loads are obtained with the engine in working order, under which condition the total weight is 98 tons 10 cwt. The driving wheels are placed 7 ft. 4½ ins. apart, so that the rigid wheel base is only 14 ft. 9 in., which factor combined with the added flexibility yielded by the bogie

at each end enables the locomotive to negotiate easily the many severe curves met with on the London, Brighton and South Coast line, and further, it makes for great smoothness of running, even when travelling at the highest speed. Other interesting dimensions are given on the drawing, among which it may be noted is the length overall, which just exceeds 50 ft.

The appearance of this engine is very handsome indeed, as will be at once admitted on a glance at the photograph reproduced on Plate VI. Although the cylinders and the valve gear are placed outside the frames, there is an entire absence of any undue complication such as is generally associated with Continental locomotives. The cab is very roomy, giving perfect protection to the driver and fireman no matter in what direction the engine may be running, and the provision of a bogie at each end enables the engine to run with equal smoothness in either direction. Despite these facts, however, the engine is nearly always run front first when hauling a passenger train, i.e. with the chimney leading, since all the controls are then at the driver's hands as he keeps his look-out in the direction the train is running. As will be seen from the drawing showing the end view of the locomotive, two sets of "blinkers" are provided, to enable the driver to keep a good view of the signals, etc., one pair at the side of the fire box and a smaller pair of circular windows above the fire box. It is recognised that with the "Belpaire" fire box, the look-out is rather impaired; but the type of fire box has its advantages, not the least of which is that with the top and the sides perfectly flat, the problem of staying and making the connections with the boiler shell becomes greatly simplified. This class of engine is more than sufficient to meet the requirements of the Brighton railway, both as regards the very heavy passenger trains and the fast expresses, which do the 50½ miles from London (Victoria) to Brighton without a stop in one hour. The famous "Southern Belle," a train consisting entirely of Pullman cars, is generally hauled by one of these engines, and while 50½ miles in 60 minutes may not seem an excessive speed, it calls for very good running in view of the many severe gradients and the long distance before London is cleared, in which high speeds cannot be attempted. The writer, who has frequently made this run in well under the hour, often indulges in the pleasure of taking a "log" of the journey, and

Fig. 148.—A View of the Front of the "Pacific" Type Locomotive, London and North-Eastern Railway.

it is no uncommon thing to find that a speed of 75 miles an hour and even more is frequently reached and maintained for stretches of three or four miles. With a load often approximating 400 tons, including the engine itself, this is a very fine performance.

In order to demonstrate the effect of the loading gauge upon the dimensions of locomotives and at the same time to give the reader some idea of colonial and foreign practice, we will now consider a few examples of locomotives used abroad. We have already referred to the fact that in certain parts of India the gauge of the railway is 5 ft. 6 in. It so happens that on the Great Indian Peninsula Railway where this gauge has been adopted the available head room is not very much different from that available in this country, viz. 13 ft. 6 in. but with the increase of width at the rails it is possible to build the rolling stock with a maximum width of 10 ft 6. in. This enables very large and powerful locomotives to be built, as will be easily understood from a study of the locomotive shown in Fig. 150, which is a reproduction from a photograph of one of a number of heavy goods engines designed and constructed for the Great Indian Peninsula Railway by the North British Locomotive Company, Ltd., of Glasgow, to whom we are indebted for the photograph and for the accompanying particulars. These locomotives have been designed for working heavy goods trains, some of which have a weight of 1,600 tons behind the tender, between Bombay and Bhusaval, where gradients of 1 in 150 are met with. On these, the locomotives work without assistance; but on sections where grades of 1 in 37 occur, special locomotives are provided to give assistance to the train locomotive. The engines are of the 2–10–0 type —the "Decapod" type as it is

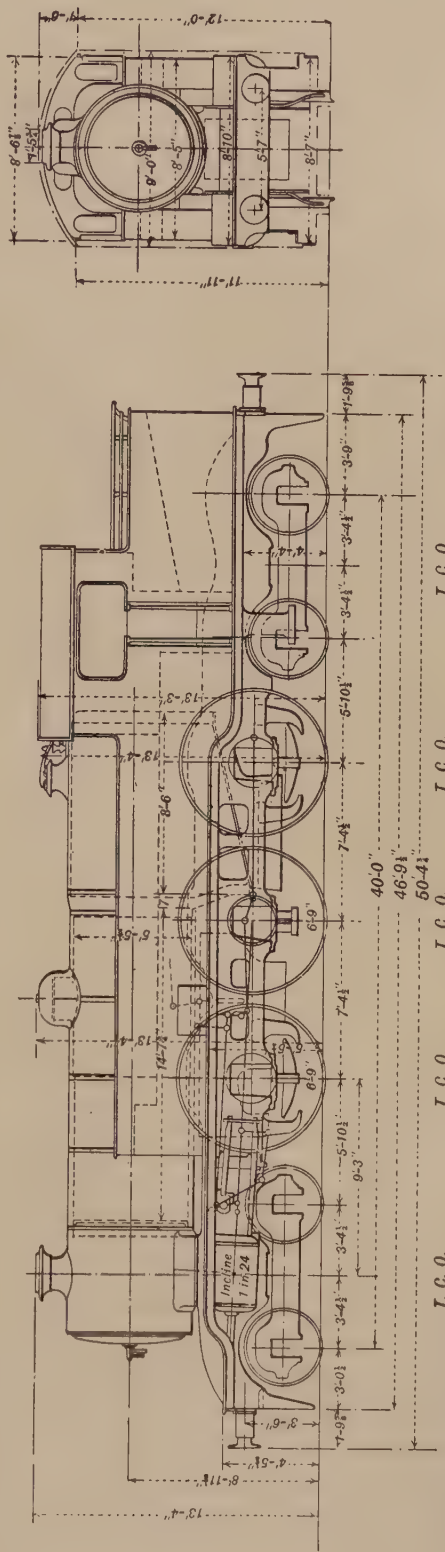

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FIG. 149.—General Arrangement Drawings of Express Tank Locomotive of the "Baltic" Type for the London, Brighton & South Coast Section of the Southern Railway.

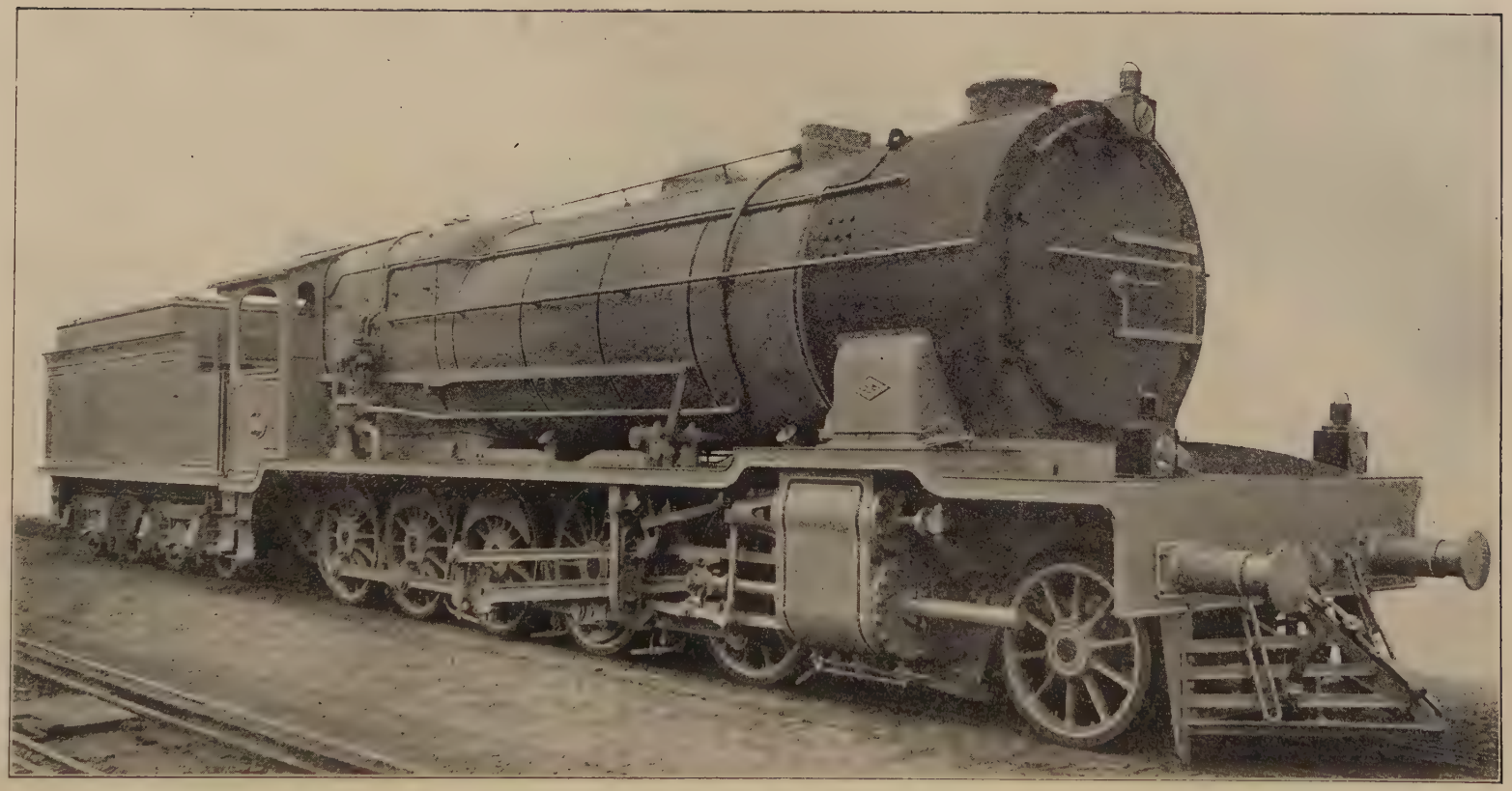

FIG. 150.—Broad Gauge Heavy Goods Engine for the Great Indian Peninsula Railway (5 ft. 6 in. gauge), designed and constructed by the North British Locomotive Co., Ltd., Glasgow.

frequently termed—there being a leading pair of wheels and five pairs of coupled wheels. The leading wheels, which are pivoted so that they can function like a bogie, are 3 ft. 7 in. diameter, while the coupled wheels are 4 ft. 8½ in. diameter.

These locomotives are equipped with four cylinders, all of which receive steam directly from the boiler, the engines being thus non-compounded. The cylinders are each 20 in. diameter with a piston stroke of 26 in., the inside pair driving the second pair of coupled wheels, while the outside cylinders drive the third pair of coupled wheels. It may be noted that the third and fourth pairs of coupled wheels are flangeless, thus facilitating the movement of the engine around the various curves which have to be negotiated. The outside cylinders are set with their centre lines horizontal, while the inner pair are raised so that they slope downward at an incline of 1 in 7¼ towards the crank axle, thus enabling the connecting rod to clear the axle of the first pair of coupled wheels. Steam is distributed to the various cylinders by means of piston valves, these being placed above the outside cylinders, as will be seen from the photograph, while they are situated below the inside cylinders. The valves are actuated by means of only two sets of Walscheart's valve gear, these being fitted outside the frames and working the valves of the outside cylinders direct, while rocking levers connected to the end of the valve rods serve to transmit the motion to the valves of the inner cylinders. The end of this rocking lever will be clearly seen in the photograph. Steam is supplied to the cylinders by a boiler of considerable capacity, having a working pressure of 160 lb. per sq. in. The diameter of the boiler is 6 ft. 9 in., its centre being pitched a distance of 9 ft. above the rails. The barrel is built up of three rings and is 16 ft. 4 in. long between the tube plates. The fire box is of the Belpaire pattern, the inside dimensions being 10 ft. 8⅜ in. long by 4 ft. 1⅜ in. wide, giving a grate area of about 45 sq. ft. The boiler itself contains 193 fire tubes each 2¼ in. external diameter, and in addition there are 36 superheater elements, the heating surface of the boiler being 2,968 sq. ft. made up as follows :

Heating surface of boiler :

	sq. ft.
Tubes (small)	1,914
Tubes (large)	824
Fire box	230
Total . .	2,968

The superheater has a surface of 617 sq. ft.

Oil fuel is used in preference to coal in view of the facilities which exist for obtaining oil, the fuel injection being operated on the steam jet system,[1] arrangements being made for coupling the burners to an outside source of steam when the locomotive is being prepared for duty and no supply from the boiler itself is available. The tender is designed to carry a large oil tank having a capacity of 9 tons of oil, this tank being so designed that it can be lifted bodily off the tender if desired. Beneath the oil-fuel tank is the water tank, which has a capacity of 5,000 gallons. A feed pump, seen in the photograph near the cab, and an injector are provided for transferring the water from the tender to the boiler, the water being delivered into the boiler through non-return valves fitted at the top of the boiler just behind the chimney. The tender is carried upon two bogies having wheels 3 ft. 7 in. diameter, and when carrying a full load of oil and water weighs 66 tons 9 cwt. The weight of the engine in full working order is 107 tons 19 cwt., of which 94 tons 10 cwt. is on the coupled wheels, there being approximately 19 tons on each pair of coupled wheels. Thus the total weight of the engine and tender in full working order is 174·43 tons. As already noted, the width of the locomotive is 10 ft. 6 in., this

[1] See Volume IV, Chapter VII, for a discussion of the various systems of oil burning.

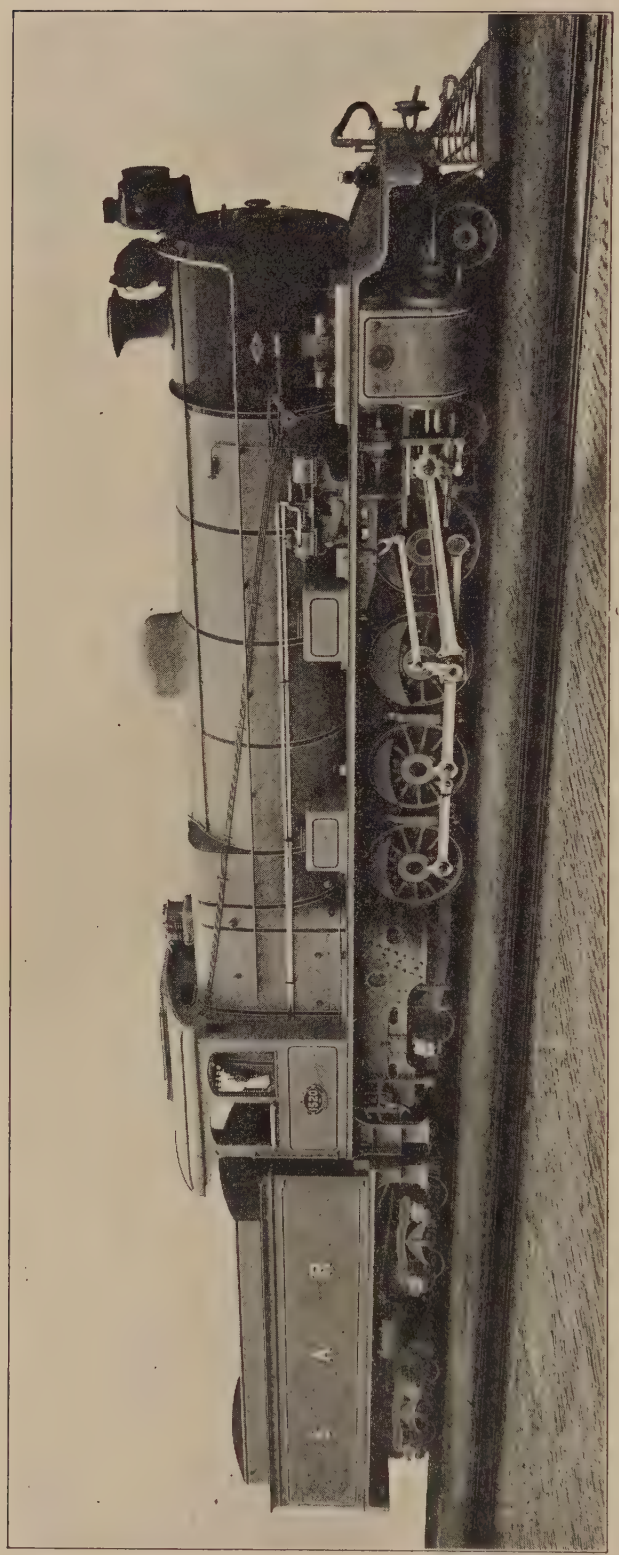

FIG. 151.—Narrow Gauge Heavy Goods Engine for the South African Railways (3 ft. 6 in. gauge), designed and constructed by the North British Locomotive Co., Ltd., Glasgow.

being the width of the buffer beam, while the total length of engine and tender measured over the buffers is 70 ft. 5 in., of which the engine is responsible for about 43 ft. It is of considerable interest to compare these dimensions with those of a locomotive built for the standard gauge of 4 ft. 8½ in. in order to determine the value of the wider gauge.

In certain of the colonies the 3 ft. 6 in. gauge has been adopted, notably in various parts of Australia and also in South Africa. In order to demonstrate the effect of the limitation of the gauge to this figure, we may consider briefly the dimensions of a heavy goods engine designed and constructed by the North British Locomotive Co., Ltd., for the South African Railways. This locomotive, which is illustrated in Fig. 151, is of the 4–8–2 type, to which no particular name has been given, has a leading bogie the wheels of which are 2 ft. 4½ in. diameter, eight coupled wheels each 4 ft. 3 in. diameter, while the trailing wheels are 2 ft. 9 in. diameter. There are two cylinders placed outside the frames—it becomes almost impossible to place the cylinders inside the frames on narrow-gauge engines unless very small cylinders are employed—these being 24 in. in diameter with a piston stroke of 26 in. The valves, which are of the piston type, are actuated

by means of Walscheart's gear. Superheated steam is supplied at a working pressure of 185 lb. per sq. in. from a boiler in which the total heating surface is 2,510 sq. ft., the fire box being responsible for 209 sq. ft. The superheater has a heating surface of 466 sq. ft., and the grate area is 40½ sq. ft. The tender, which is carried on two bogies having wheels 2 ft. 9½ in. diameter, has a water capacity of 4,250 gallons, while the space available for the carriage of fuel is 450 cu. ft., which, assuming that 1 ton of coal occupies 43 cu. ft., gives the coal capacity as 10½ tons. The weight of the engine in full working order is 96 tons 18 cwt., of which 69 tons 3 cwt. is carried by the coupled wheels, thus yielding an average of 17 tons 5¾ cwt. on each pair of coupled wheels. The weight of the tender when in full working order is 51 tons 2 cwt., so that the total weight of engine and tender is 148 tons. The total wheel base of the engine, i.e. the distance from the centre of the leading pair of bogie wheels to the centre of the trailing wheels, is 32 ft. 1 in. The wheel base of the tender is 16 ft. 9 in., and the total wheel base—that is, the distance from the leading pair of wheels on the engine to the last pair on the tender—is 58 ft. 7¼ in.

Our survey of the various types of locomotives would not be complete without a brief reference to those which are built in the United States for service on the American railways. Although the standard gauge, viz. 4 ft. 8½ in., has been adopted in America the loading gauge is much more generous, the available distance from the track to the top of the engine being no less than 15 ft. with a correspondingly large extreme width. In addition to this we find that it is the American practice to employ trains of very great weight, particularly in the goods service (freight trains, as our American cousins call them), and as a result the largest and most powerful in the world are to be found on some of the American railways. While, however, the size of the locomotives is considerably in excess of anything to be found in this country, it may be noted that neither in average speed nor in the length of the non-stop runs can American practice claim to be superior to our own, although perhaps as regards the former there is little to choose, as in both countries the average of 60 miles per hour is regarded as the ultimate goal to be achieved.

In order to give the reader as good an idea of American locomotive practice as is possible in the space at our command, we will consider first of all an engine designed and built for hauling express passenger trains and we will then deal with an example of the famous freight locomotives to be found in the United States. We are indebted to Messrs. The Baldwin Locomotive Works, of Philadelphia, one of the largest and best-known firms of locomotive builders in the world, for placing at our disposal illustrations and particulars of a large number of locomotives built by them from which we have made a selection, typical of the various types of engine employed at the present time in America. It may be noted in passing that while in appearance the American-built engine appears very different from that with which we are familiar in this country, the fundamental principles underlying the design are in each case the same, as also is the general lay-out. Thus, we have the same type of boiler, we have the cylinders arranged at the front of the frame and exhausting into the smoke box, although it may be mentioned that in the heaviest and most powerful engines a second set of cylinders is sometimes employed, which drives a second set of coupled wheels placed beneath the back end of the boiler. Then, we find that the cab is in precisely the same position with the tender coupled to the engine, thus enabling the fireman to have ready access to the fuel and the water. Of course, there have been a few "freak" designs, in which the position of everything has been altered, cab at the front or in the middle of the boiler, and so on, but, generally speaking, the designs adopted reveal the interesting fact that both in the United States and in this country locomotive practice has evolved from a common origin and that it has developed along parallel lines, thus, it might be added, indicating that the design evolved is the one which is best suited to meet the conditions of actual service.

As an example of a locomotive built for the express passenger service on an American

railway, we may consider the " Pacific " type of engine shown in Fig. 152. This engine was built by the Baldwin Locomotive Works for the Chicago, Burlington and Quincy Railroad (Fort Worth and Denver City Railway). The driving wheels are 6 ft. 2 in. diameter, the leading bogie has wheels 3 ft. 1¼ in. diameter, while the trailing wheels are 4 ft. 0¼ in. diameter. The locomotive has two cylinders placed outside the frames driving on to the second pair of coupled wheels, the distribution of the steam being by piston valves mounted on the top of the cylinders and operated by Walscheart's valve gear. The cylinders are 27 in. diameter with a piston stroke of 28 in., while the piston valves are 14 in. diameter. Steam is supplied at a pressure of 180 lb. per sq. in. from a boiler of the " wagon " type, which is virtually a double conical boiler tapering towards the cab as well as towards the smoke box, the former being in order to facilitate the look-out from the driver's platform in the cab. At its largest part, the diameter of the boiler is 6 ft. 6 in., and the length between the tube plates is 18 ft. 6 in. The fire box, which,

FIG. 152.—Express Passenger Locomotive of the " Pacific " Type built by the Baldwin Locomotive Works, Philadelphia, for the Chicago, Burlington and Quincy Railroad.

like the boiler, is made of steel, is of the " Wootten " type, 9 ft. 0⅛ in. long, and 6 ft. 6¼ in. wide at the bottom, having a depth of 7 ft. 1¾ in. at the front and 6 ft. 0 in. at the back. The boiler is fitted with 200 tubes each 2¼ in. diameter, while there are 34 superheater tubes each 5½ in. diameter. The total heating surface of the boiler is 3,364 sq. ft., of which the fire box contributes 233 sq. ft., while the superheater has a heating surface of 751 sq. ft. Oil fuel is burned in the fire box, which has a grate area of 58·7 sq. ft. The tender is carried on two bogies having wheels 3 ft. diameter and has a capacity for 8,200 United States gallons of water and for 3,200 U.S. gallons of oil.[1]

The rigid wheel base of the engine, i.e. the distance between the first and last pairs of wheels between which no play or side movement, such as is provided by bogies, is possible, is only 13 ft., which means that the engine would be able to negotiate curves with considerable smoothness. The rigid wheel base, it may be noted, is generally the distance between the centres of the front and back driving axles, although in large

[1] The United States gallon is smaller than the British Imperial gallon, the former containing 23 cu. in. and the latter 27·727 cu. in. Hence to convert U.S. gallons to British gallons divide by 1·2.

freight locomotives where the number of pairs of coupled wheels often exceeds four, a certain degree of flexibility between the driving wheels is often provided. The total wheel base of the engine under consideration is 33 ft. 8½ in., and including the tender it is 65 ft. $10\frac{3}{16}$ in. The total weight of the engine is 292,750 lb., i.e. 146·4 United States tons, or 130·6 British tons,[1] of which 179,660 lb. is carried by the driving wheels. It is interesting to note the 179,600 lb. on all the driving wheels is equivalent to 59,866 lb. or 26·7 British tons on each driving axle, which is considerably greater than the permissible load on any single axle of a British locomotive. The greater permissible load on any axle is another reason which accounts for the greater dimensions, etc., of American locomotives as compared with those in this country. The total weight of the engine and tender is 459,000 lb., i.e. 205 British tons.

There are one or two constructional features of this locomotive which call for comment as being indicative of American practice. The first is the frame. This is not built up of

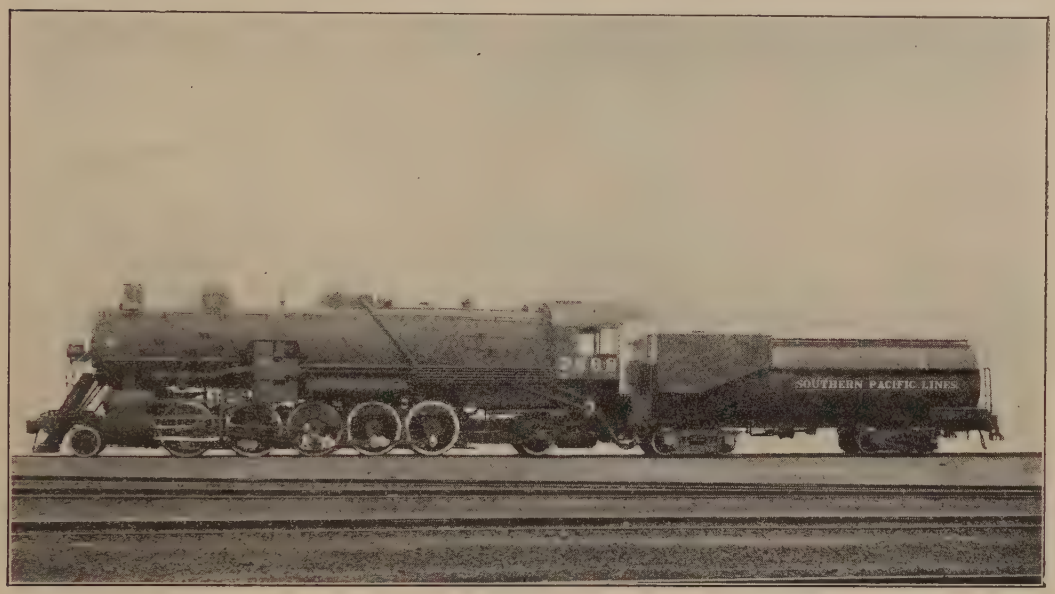

FIG. 153.—A large " Freight " Locomotive built by the Baldwin Locomotive Works for the Southern Pacific Railway.

two parallel plates in the manner always adopted by British designers, but is composed of a number of bars to which the cylinders and the driving wheels are secured and upon which the boiler is mounted. This use of the " bar frame " instead of the " plate frame " is a feature of American locomotive construction and accounts for the extremely open appearance of the American-built locomotive. The familiar " cow catcher " in the front of the engine and the warning bell mounted on top of the boiler may be noted. Again, the large dome bearing the number may be noted. This is the sand box from which small pipes run down to the front of the first pair and to the rear of the second pair of driving wheels, so that sand can be applied when the rails are wet. The large feed pump mounted directly over the middle pair of driving wheels also may be noted. Immediately behind this is the feed water heater, while the steam for operating the pump is taken from a connection mounted on the top of the boiler immediately in front of the cab. Other

[1] In the United States two values of the ton are employed, the " short ton " = 2,000 lb. and the " long ton " = 2,240 tons. The former is sometimes known as the United States or American ton, while the latter is, of course, the British ton.

details might be mentioned, but the reader should be able, by a careful study of the photograph reproduced in Fig. 152, to determine the function of the various pipes and fittings exposed to view. He will then recognise that the majority of these are to be found on all British locomotives, but it is usual to fit them in position where they are not so readily seen. We are not going to venture an opinion as to which arrangement is to be preferred, neither do we consider it necessary to discuss the problem of the relative merits of British-built and American-built locomotives. Both, it may be stated, are excellently suited for the peculiar duties which they are called upon to perform, and that, after all, is the main desideratum.

Considering next an example of a large American freight locomotive, that shown in Fig. 153 may be taken as typical of the type which is very extensively employed on the railroads of the United States. The locomotive shown has been built for the Southern Pacific Railway Company by the Baldwin Locomotive Works. It is of the 2–10–2 type (another type to which no particular name has been given), the driving wheels being 5 ft. 3½ in. diameter. The front wheels, which are so mounted that side play is permitted, are 2 ft. 9 in. diameter, while the rear wheels similarly mounted are 3 ft. 9½ in. diameter. There are two cylinders, each 29½ in. diameter, with a piston stroke of 32 in., the distribution of the steam to the cylinders being carried out by means of piston valves 15 in. diameter, which are mounted above the cylinder, the valves being actuated by Walscheart's gear. The cylinders and valve chests are outside the frames, this appearing to be the general practice in America. The boiler is of the straight-top type, having a diameter of 7 ft. 6 in., while the length between the tube plates is 21 ft. The fire box is 11 ft. long, 7 ft. 6 in. wide at the base, the depth at the front being 7 ft. 4¾ in. and at the back 6 ft. 1 in. The fire box, which is constructed of steel, is of the Wootten type —a type which is very widely employed in America—the grate area being 82·5 sq. ft. The boiler contains 261 fire tubes, each 2¼ in. diameter, and 50 superheater tubes, these being 5½ in. diameter. The total heating surface of the boiler is 5,103 sq. ft., of which the fire box contributes 251 sq. ft., and to this total must be added an additional 1,329 sq. ft. provided by the superheater. The working pressure is 200 lb. per sq. in., and the boiler is equipped for the burning of oil fuel. The tender, which is of cylindrical form—a form not so far employed in this country—is carried on two bogies, the diameter of the wheels being 2 ft. 9 in. The water tank has a capacity for 12,000 U.S. gallons, and superimposed above this is the oil tank, the capacity of which is 4,000 U.S. gallons.

The total wheel base of this locomotive is 82 ft. 7½ in., of which the engine is responsible for 42 ft. 4 in., the rigid wheel base, which in this case is the same as the driving wheel base, being 22 ft. 10 in. The weight of the engine and tender is 621,000 lb. (277 British tons), and the weight of the engine alone is 398,000 lb. (178 British tons). Of this weight 306,000 lb. (136·5 British tons) comes on the driving wheels, which is equivalent to 27·3 tons on each driving axle. The general features of the design follow very much along the lines of the express passenger locomotive described above.

There remains one other type of locomotive to which attention may be drawn. Such a locomotive is shown in Fig. 154, and while at first appearance it might be regarded as a freight locomotive, actually it has been designed for what is termed in America "switching" service—that is, for shunting or marshalling the trains in large goods yards. The locomotive under notice has been built by the Baldwin Locomotive Works for the Cambria Steel Company, and in view of the overall length, which is not excessive, it will be realised from the particulars appended that she is a very powerful locomotive indeed, being particularly suited for the difficult duty of shunting, with its frequent starts and stops when dealing with heavy loads. The engine is designed for the standard gauge, 4 ft. 8½ in., and is of the 0–8–0 type, having driving wheels 4 ft. 8 in. diameter.

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PLATE VI.—EXPRESS TANK LOCOMOTIVE OF THE "BALTIC" TYPE FOR THE LONDON, BRIGHTON AND SOUTH COAST SECTION OF THE SOUTHERN RAILWAY.

There are two cylinders, each 27 in. diameter, with a piston stroke of 30 in. The distribution of the steam to the cylinders is carried out by piston valves, each 14 in. diameter, which are mounted above the cylinders and actuated by Walscheart's valve gear. Steam is supplied at a pressure of 205 lb. per sq. in. from a boiler of the straight top type, the diameter of which is 7 ft. 2 in., while the length between the tube plates is 14 ft. 11 in. The fire box of the usual American type has a length of 9 ft. 6 in. and a width of 6 ft. $0\frac{1}{4}$ in., the depth at the front and the back being 6 ft. $5\frac{1}{4}$ in. and 5 ft. $2\frac{3}{8}$ in. respectively. The boiler contains 237 fire tubes each 2 in. diameter, and 40 superheater tubes each $5\frac{1}{2}$ in. diameter. The total heating surface of the boiler is 2,948 sq. ft., of which the fire box yields 211 sq. ft., while the superheater is responsible for a further 750 sq. ft. of heating surface. The boiler is designed to burn soft coal, the grate area being 57 sq. ft. The tender is carried on two bogies, the wheels of which are 2 ft.

FIG. 154.—Shunting Locomotive built by the Baldwin Locomotive Works for the Cambria Steel Co., U.S.A.

9 in. diameter, and the water capacity is 8,000 U.S. gallons, while provision is made for the carriage of 14 tons of fuel.

The wheel base of the engine, which is also the rigid wheel base, is 16 ft., while the total wheel base of the engine and tender is 52 ft. 2 in. The engine in full working order weighs 270,000 lb. (i.e. 120 British tons), and since all this is carried by the driving wheels the weight on each driving axle is 30 tons. The total weight of engine and tender is 425,000 lb. or approximately 190 British tons. The feature of this engine is the weight carried by each pair of driving wheels, which is made large in order to facilitate rapid starting when dealing with heavy loads. It will be noticed that two sand boxes are provided, one to the front wheels and one to the rear wheels, and that a steam cylinder is fitted for operating the reversing gear. Altogether, the locomotive appears to be particularly suited to fulfil the heavy duties which it is called upon to perform.

It would, of course, be possible to say a great deal more on the fascinating subject of locomotives. It is undoubtedly true that to many people the locomotive makes an appeal far greater than that which is exercised by any other type of engine. The reason, may, perhaps be found in the fact that very many people have an opportunity of seeing

locomotives at close quarters, while access to other forms of power generators is denied them, or it may be that there is something extraordinarily imposing in the sense of power and speed which is embodied in the small, sleek-looking machine that stands harnessed at the front of a long train with only a slight wisp of steam escaping from the valves to indicate the mighty forces which are chained up within it. Be that as it may, to many the locomotive is the only example of the wonders of engineering that they have an opportunity of seeing. They read in their newspapers of mighty ships and of wonderful machinery, but any further acquaintance with these triumphs of man's engineering skill is denied them. Thus their awe and their admiration of the railway locomotive is increased. Their sons, as boys enjoying a ramble or a holiday with their father, gradually feel and ultimately share this enthusiasm, and in very many instances it is the love of locomotives thus engendered which leads them to adopt engineering as a career in after-life. This love—toned down, perhaps, by the passage of years—remains, despite the fact that circumstances may have urged them towards some other branch of engineering.

FIG. 155.—The "Sentinel" 4–5 ton Steam Wagon, designed and constructed by the "Sentinel" Wagon Works (1920), Ltd., Shrewsbury.

Thus the boy who longed to become "an engine driver"—and what real boy did not?—becomes an engineer on board ship or in an electric power station, but his interest in locomotives survives. If there are any such who read these pages—and the author is convinced that there will be many such—they will perhaps be disappointed because space has not permitted us to describe other types of steam locomotives. To these we can only repeat that there is an extensive literature devoted exclusively to the subject of the locomotive, and this must be consulted if further information is desired. To the others, the engineers who view the problem of the generation of power as a whole or whose enthusiasm, perhaps, is towards some other form of machinery, it is hoped that the above brief notes will indicate the general principles underlying the design of the steam locomotive, giving a good idea of the powers which it can develop and demonstrating its relationship to other forms of power generators.

Before concluding this chapter, reference must be made to a form of locomotive which has within recent years become of very great importance in connection with heavy road transport. We refer, of course, to the steam wagon. It is a matter of historical interest that some of the earliest attempts to employ the steam engine for purposes of transport on land were made with road vehicles. Richard Trevithick has been awarded the honour of having invented the first steam locomotive to run on a railroad, his famous engine for the Pen-y-darran railroad in Wales having been built in 1804. But Trevithick himself had made a road vehicle driven by a steam engine in 1802, while many other inventors were tackling the same problem even earlier than this. It can in fact be said that the

first applications of the steam engine to the propulsion of vehicles on land were made, not on railway locomotives but on road wagons. These early attempts were not, however, attended with any great measure of success. On the other hand the steam locomotive,

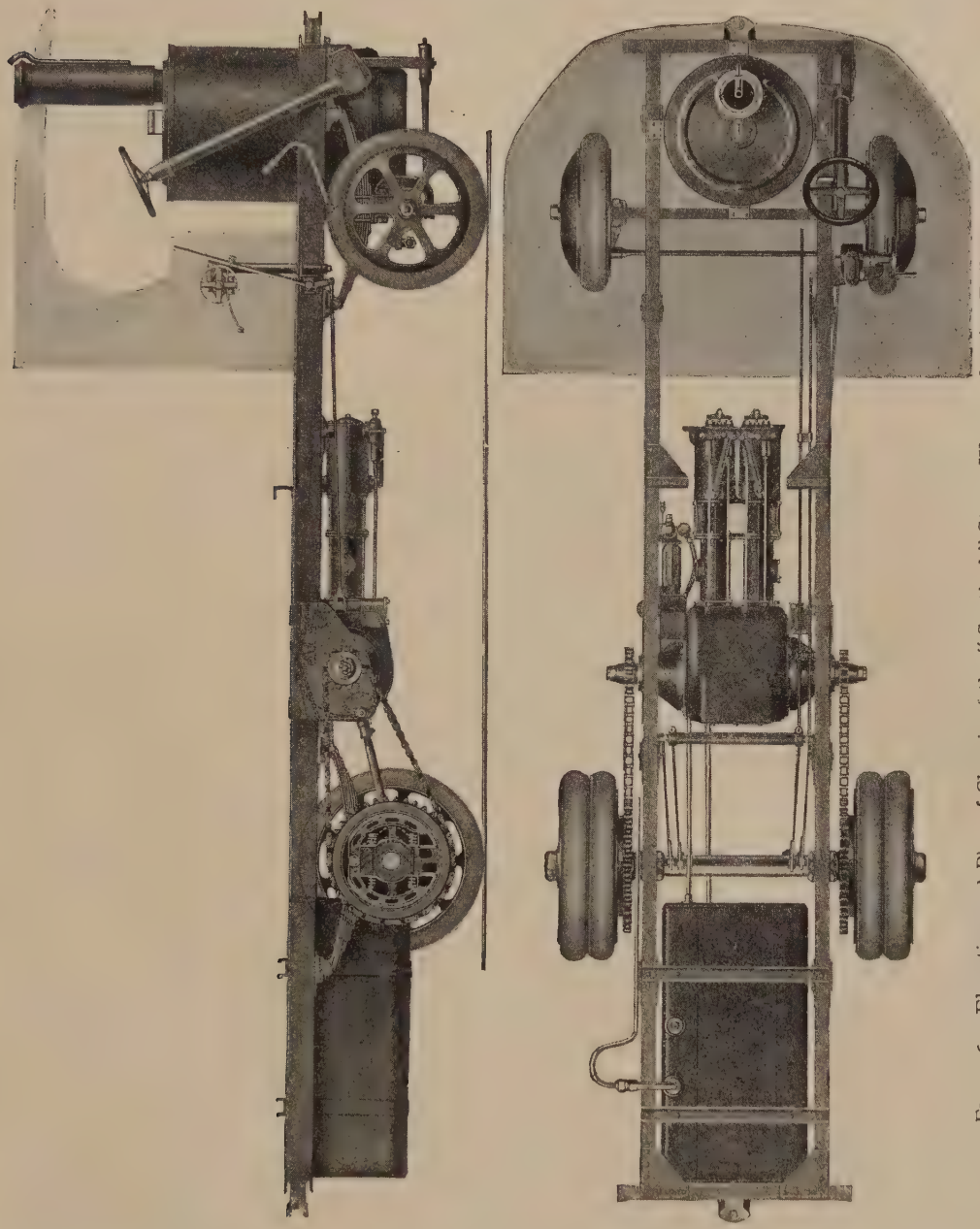

FIG. 156.—Elevation and Plan of Chassis of the "Sentinel" Steam Wagon, showing Boiler and Engine.

particularly after the appearance of Stephenson's *Rocket*, was developed with great rapidity, and thus the minds of engineers were gradually drawn away from the possibilities of road transport, and the steam wagon was for a long time regarded as being of no practical value. With the introduction of the motor car towards the end of last century and the application of the petrol engine to the propulsion of heavy road vehicles used for the

transport of merchandise, the possibilities of the steam engine were again brought forward, and with the advance that has been made, not only in the design and construction of the steam engine and its boiler, but in the design of the "chassis" upon which these are mounted, the problem which more than a century ago baffled our forefathers has to-day been successfully solved.

There are a number of different forms of steam wagons on the market at the present time, many of which can rightly be described as locomotives in miniature, possessing a small boiler of the locomotive type, and propelled by an engine which is in many respects very similar to the engine of a locomotive but, of course, built on a smaller scale. In certain designs, however, a bold departure has been made from previous practice, both as regards the design of the boiler and of the engine, and it will be of greater interest if we consider in detail a steam wagon to the design of which the above statement can be applied. One of the best-known steam wagons, and certainly one of the most successful, is the "Sentinel," which is made in a number of standard sizes by the "Sentinel" Wagon Works (1920), Ltd., of Shrewsbury. The reader has doubtless seen many of these "Sentinel" steam wagons on the road, but in order to give a general idea of the appearance of this type attention may be drawn to Fig. 155, which shows the makers 4–5 ton model, the name "Super-Sentinel" having been given to the firm's latest models in view of the many improvements introduced into the design of the earlier "Sentinel" wagon.

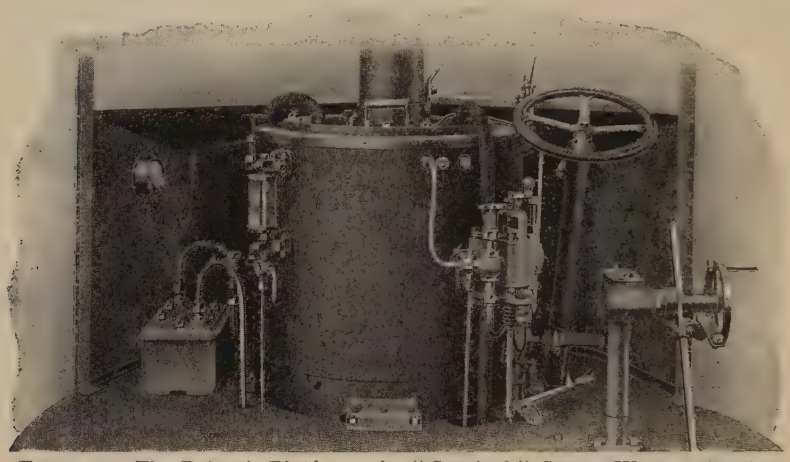

FIG. 157.—The Driver's Platform of a "Sentinel" Steam Wagon, showing the Steering Gear and the various Controls.

The wagon is designed to carry a load of from 4 to 5 tons, at a service speed of 12 miles per hour. The platform upon which the goods are carried can be either 13 ft. 6 in. or 15 ft. 3 in. in length, the corresponding wheel base being 10 ft. 9½ in. or 12 ft. 6½ in., while the standard width of the platform is 7 ft. The tyres on the front wheels are 900 mm. diameter by 180 mm. wide, while on each back wheel there are two tyres 950 mm. diameter by 160 mm. The actual form of the body which is fitted depends, of course, on the nature of the loads which are to be carried, these wagons often being employed for some specific purpose, rather than for the transportation of general merchandise. In this book, however, we are not concerned so much with the body as with the chassis and the machinery which is employed to propel the wagon.

The general lay-out of the machinery of a "Sentinel" steam wagon is given in Fig. 156, in which is shown the elevation and the plan of the chassis, complete with all its equipment. It will be noticed that steam is provided by a vertical cylindrical boiler mounted between the frames at the front of the vehicle, while the engine is carried between the frames at the middle of the chassis, the engine shaft driving the rear wheels by means of roller chains. The frame consists of two straight steel beams of channel section, kept together, both at the front and at the back as well as intermediately, by a number of cross beams, also of channel section. The frame is carried on the front and rear axles

by means of strong semi-elliptical springs as shown. The boiler is carried low down between the frames, so that the driver obtains a very clear view of the road in front of him, while a comfortable and roomy cab gives protection against the weather. All the controls are within easy reach of the driver, as will be seen from Fig. 157, which is a view of the driver's platform taken from just behind the seat. For the control of the engine itself there is only one lever, shown on the right, which is pushed forward to go ahead, pulled back to reverse and left in the mid position to stop, in which position all the valves are held open to drain the cylinders. The speed of the vehicle is regulated by the amount of steam passing through the stop valve, which is operated by means of the small hand wheel adjacent to the control lever. A steam pressure gauge and a water gauge are provided on the left, but in full view of the driver, and the injector which feeds the water into the boiler is just in front of the driver. A feed pump is also provided, this being driven directly off the engine, so that the injector is available as an alternative to the pump or as a stand-by, while a valve is fitted within easy reach of the driver for by-passing the water from the feed pump back to the tank if the level in the boiler tends to get too high. The fuel is fed into the boiler through a small door on the top of the boiler, gas coke being the fuel recommended, although, of course, coal or anthracite can be burned. In addition to the foot and hand brakes, which act on the wheels in the usual way, a patent engine brake is provided for use on hills and when manœuvring. A foot pedal is pressed down which shuts off the supply of steam to the engine. If now the engine is reversed and the foot pedal is allowed to rise, the engine acts as a compressor, and according to the pressure on the foot pedal it retards the wagon, forming an immensely powerful brake, which effectively holds the wagon on the longest hills without the heating which is inseparable from the continued use of friction brakes. The steering wheel of the usual form completes the equipment in the cab. It may be noted that while one man can handle the wagon with ease, a boy is usually carried to stoke the boiler and to assist in loading and unloading.

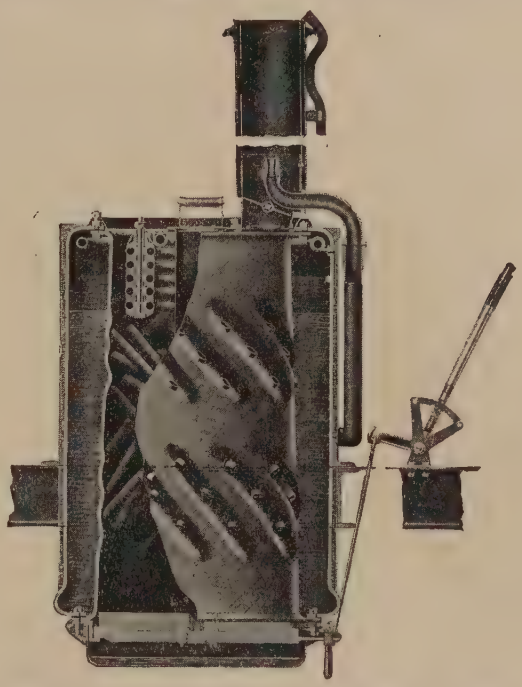

FIG. 158.—Sectional View of the Boiler of a "Sentinel" Steam Wagon, showing Fire Box, Water Tubes, and Superheater.

The general features of the design of the boiler of the "Sentinel" steam wagon will be readily understood from a study of Fig. 158. It will be seen that the boiler is of the water tube type, the central cylindrical fire box being provided with a number of straight solid drawn steel tubes, which are arranged in a sloping position across it. These tubes are expanded into the fire box shell at a number of landings, so that they can lay tangential to the stoking hole without any curvature being given to the ends of the tubes. The whole of the fire box with its tubes can be readily removed from the drum of the boiler for cleaning purposes by undoing two rings of bolts, one at the top and one at the bottom, the fire box being made slightly conical to facilitate its removal from underneath. This enables the interior of the water spaces to be cleaned also. A steam jet is also provided

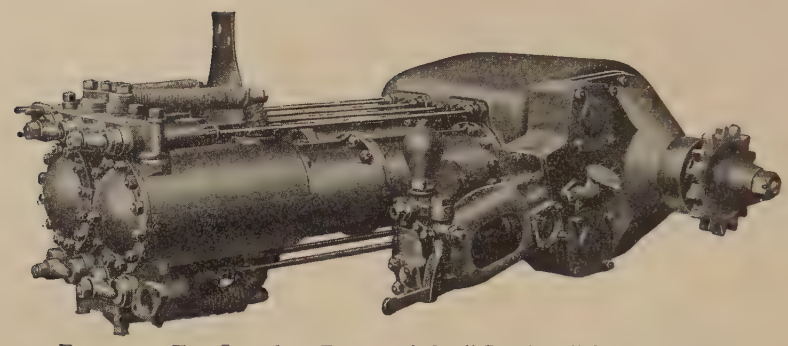

FIG. 159.—The Complete Engine of the " Sentinel " Steam Wagon.

for removing soot from the tube surfaces when the wagon is on duty. It will be noticed that the exhaust from the engine is led to the chimney, thus providing an efficient draught to the fire and enabling coke to be readily burned, while for raising steam or for maintaining the fire when the engine is not running, a small blast pipe is provided at the top of the funnel as shown. The ashes from the fire box fall into an ash tray, which carries a small quantity of water and an arrangement is fitted for tilting the fire grate to facilitate the removal of ash and clinker. It will be noticed that at the top of the fire box a superheater is provided, consisting of a number of coils of tube of small diameter. The use of superheated steam adds materially to the efficiency of the engine. It may be noted that the working pressure of these boilers is 230 lb. per sq. in., and that they are hydraulically tested to 350 lb. per sq. in. before leaving the makers' works.

The engine of the " Sentinel " wagon is a very fine piece of mechanism. The general appearance of the engine as fitted in the chassis will be understood from Fig. 159, from which it will be noticed that it is of the two-cylinder simple non-condensing type, the cylinders being of the same diameter, and that the whole is totally enclosed to protect it from dust and mud, the feed pump being arranged at the side, also totally enclosed. Sectional views of the engine are reproduced in Fig. 160, from which the principal features of the design can be gathered. As in locomotive practice, the two cranks

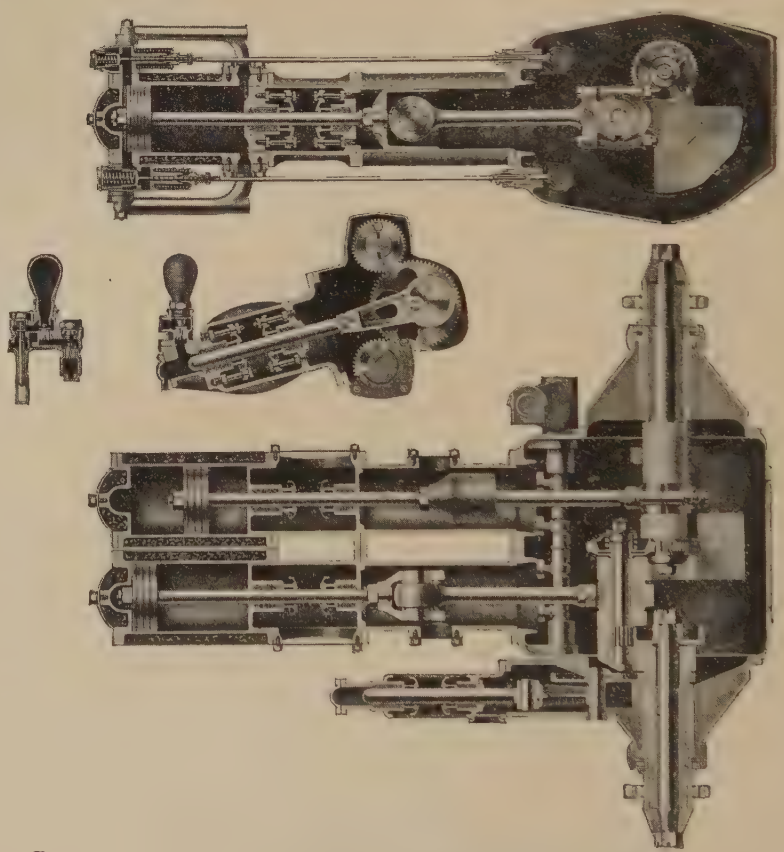

FIG. 160.—Sectional Elevation and Sectional Plan of the Engine of the " Sentinel " Steam Wagon.

are arranged at right angles to enable the engine to start up in any position, but here it will be noticed a differential gear is provided, so that one sprocket can be driven at a different speed to the other, as when rounding corners, etc. The fitting of the differential gear in the crank case instead of at the back axle is of considerable advantage, as it not only enables it to be removed out of the dust and mud to a position where it can be properly lubricated, but it enables a stationary back axle to be used. It will be noticed further that the valves are of the mushroom type driven by means of cam shafts, thus again differing substantially from locomotive practice. The valves are placed at the front end of the cylinders, where they are readily accessible for regrinding when necessary. While the cam shafts are necessarily driven at the same speed as the main crank shaft, the feed pump, a section of which is shown between the elevation and the plan of the main engine, is driven at a reduced speed which prevents water hammer at high speeds. The working parts of the engine—crank shaft bearings, crossheads, gears, etc.—are run in oil, which ensures perfect lubrication, and in order to prevent the oil getting into the cylinders, and at the same time to prevent steam and water getting into the crank case, double glands are fitted to the piston rods and the pump rod as shown. Provision is, of course, made for lubricating the internal surfaces of the cylinders and the valve chests. It may be noted that after a test of over 80,000 miles the engine of one of these wagons was opened up and no appreciable wear could be discovered at any of the various bearings.

When the wagon is running at a speed of 12 miles per hour the engine makes only 240 revs. per min., this slow speed obviously making for greater durability. Under normal conditions the output of the engine is about 30 h.p., although for short periods it is claimed that the engine will develop 70 h.p. The bunker capacity is sufficient to carry the wagon on a journey of about 200 miles, while the feed water tank, which is mounted right at the back of the chassis, carries enough water for a journey of 30 to 50 miles, and by a simple arrangement of the suction pipes from the tank the driver receives a warning five miles or so before the supply of water is exhausted. The fuel consumption is estimated at about 1 cwt. of coal for each 25 miles with a 4-ton load, 22 miles per cwt. with a 6-ton, and 15 miles per cwt. when fully loaded and hauling a trailer with an additional load of 4 tons. Assuming that good steam coal is obtainable at about 37*s.* per ton, coke of good quality gives the same fuel cost per mile if it is obtainable at about 27*s.* to 28*s.* per ton, and the makers do not consider it advisable to use coke if the relative cost is greater than this. There is no need for us to discuss here the relative merits and demerits of the steam wagon as compared with the petrol-driven lorry, nor are we concerned with the problem of the advantages and disadvantages of road transport as compared with rail transport. Suffice it to say that it has been thoroughly demonstrated that the steam engine driven lorry yields a highly economical and expeditious method of transferring merchandise from one locality to another, particularly if the distances involved are not excessive, and that the " Sentinel " steam wagon is both from the mechanical and the financial point of view one of the most successful types that has yet been devised.

CHAPTER VII

WATER POWER—HOW IT IS HARNESSED—WATER TURBINES—SOME MODERN EXAMPLES OF HYDRO-ELECTRIC POWER PLANTS

IN dealing with the subject of water power we are dealing with a branch of engineering which, while it is still in its infancy, is probably destined to become of vital importance. With all other forms of prime movers, steam, gas, and oil engines, we have to employ either coal or oil or some derivative from these as fuel; and thus we are dependent upon the mineral resources of the earth, which in the course of time will become exhausted. Geology teaches us that these minerals—coal and oil—have taken millions of years to form, but so rapid has been the industrial development of the world in recent times that we are now using up these resources with very great rapidity and their probable life is measured only in hundreds of years. Indeed, it is estimated that if we continue to use up our resources of coal and oil at the rate at which they are being used at present, and provided that no further considerable coal and oil fields are discovered, our coal will become exhausted in about 300 years, while oil supplies will become exhausted in about sixty years. Attempts are being made to obtain fuel from vegetable and other natural sources. Thus, reference has already been made to the use of vegetation in the generator of a gas producer, and wood, which is frequently used in this way, furnishes another example of the use of natural products for the production of power. Again, alcohol is obtainable from vegetable products, and this forms a suitable fuel for certain kinds of internal combustion engines in lieu of petrol, which is a mineral product.[1] It has been suggested that the production of these and other vegetable products might be developed upon an extensive scale in order to appreciably lessen the demand for mineral fuels, although whether this will ever be done time alone can say. But even when power is developed from a fuel derived from vegetable sources, it is necessary to first of all obtain the fuel, and this may involve a vast amount of labour and expense.

When, however, we consider the vast stores of energy contained in the waterfall and in the swiftly moving stream, or if we contemplate the incessant expenditure of energy in the diurnal flow and ebb of the tide, then we realise that Nature in her bounty has given us an abundant and never-failing source of power if only we can find ways and means to harness this power and convert it into a form better suited to the needs of mankind. There is no question here of the stores of energy being used up; no matter how we may deflect the course of the waterfall, no matter how we may steal from the tide a poor fraction of its power, they will continue to flow, ever offering us anew the gift of their ceaseless energy. In talking of water power one almost instinctively turns to the world-famous Niagara Falls. It is unthinkable that the mighty river which bears the huge volumes of water to their final plunge should ever run dry. The falls themselves may disappear, indeed it has been asserted that if the construction of hydro-electric power stations in the vicinity continues, then the falls themselves will disappear; but

[1] See Volume IV, Chapters V, VII, and VIII, for a detailed study of the subject of fuels.

the millions of gallons which to-day tumble uselessly in space will have been deflected through the water turbines and there made to drive huge electric generators, thus producing energy in a form more suitable to meet the needs of mankind. Similarly with less pretentious waterfalls, or wherever we have a watercourse which rapidly changes its level, it is possible to pass the water through a turbine which in turn is made to drive an electric generator, thereby converting its energy into a more usable form.

The use of water power on an extended scale has only become possible with the development of the methods of generating, distributing, and utilising electrical energy. Previous to that, it was essential that the power of the waterfall or the swiftly moving river—if, indeed, it were harnessed at all—should be utilised on the spot. Thus we have the familiar waterwheel at the old-fashioned farm of our country jaunts, which drives the grinding machinery whereby the corn is converted into flour. But except by the use of shafts or belts, it is impossible to transmit the power anywhere but to the immediate vicinity of the waterwheel. Less picturesque, but nevertheless examples of the use of water power, are the waterwheels—most of them now lying idle if not actually removed—which drove the cotton-spinning machinery, the looms, etc., in the weaving districts of the North Country. It is generally recognised that the cotton and woollen goods industry developed in this part of England because of the fact that it was possible to obtain water power for driving the machinery. But the industry grew with such rapidity that the demand for power outstripped the available supply. Happily the steam engine became available at the critical time. The fact that it could be used in any locality, whereas the waterwheel was of necessity chained to the stream which drove it, meant that there was no longer any restriction as to the site at which the machinery could be laid down. Moreover, the gradual introduction of steam engines of large power—large, that is, compared with the power of the majority of waterwheels—led to the gradual supersession of the latter by the former. A further disadvantage under which the waterwheel suffered was that it could only be employed in hilly districts where waterfalls and rapid streams are met with. On the plains the rivers move sluggishly forward and thus cannot be employed for operating waterwheels. Hence the manufactory and the homes of the workers had to lie in the hilly districts. This greatly increased the difficulty of expansion and also rendered difficult the problems of transport.

With the coming of electricity, however, all this was changed. Instead of locating the works in the immediate vicinity of the waterwheels, it has become possible to transmit the power through cables, often supported conveniently on standards, running through vast tracts of wild and barren country to the town or towns situated in the plains where it is used. Further, we are no longer restricted as to the site where our water power machinery shall be laid down. This can be placed wherever the energy of the stream is easy to win, far away in the mountain fastnesses in almost inaccessible districts where certainly no industry, with its homes for the workers, could ever hope to thrive. But as soon as it was demonstrated that electricity could be generated from water power and then transmitted over long distances to the large industrial centres, the science of water power engineering advanced by leaps and bounds. Instead of employing slow-speed and inefficient waterwheels, we now have efficient forms of water turbines, either direct coupled to electric generators or driving them through gearing which, as regards power, compactness, and reliability, far surpass anything which was ever attempted before the advent of electricity. Moreover, instead of being dependent upon the seasons for the amount of power which we can derive from our streams, methods have been devised for storing the excess of water which is available during the rainy season so that it can be utilised during the dry season, and thus a continuity of supply of electricity is assured throughout the year.

In order to indicate the almost phenomenally rapid developments which have occurred

in the practice of hydro-electric engineering, it may be mentioned that it was as recently as 1895 that the first water power installation was laid down at the Niagara Falls, this being also the first installation in the American continent. To-day, according to the latest report[1] of the Hydro-electric Power Commission of the Province of Ontario, the total power generated on the Canadian side alone is 729,348 h.p., while the output for the year under review reached 2,293,447,600 kilowatt hours. Figures for the Canadian side can be readily given, since all the various power stations, totalling about twenty-nine, are under the control of the one authority. The writer has not been able to obtain corresponding figures for the power generated on the American side of the falls, but these no doubt would be equally arresting. Attention may be drawn to the photograph reproduced on Plate I in Volume I of this work, which shows how the water of the Niagara River is tapped above the falls, and how it is led through canals and pipe lines to the lower level beyond the falls where it is passed through the water turbines and again discharged into the river. Each additional power station laid down will drain a certain volume of water away from the falls—incidentally requiring longer waterways and pipe lines—and thus it is easy to realise that if the process of building further power stations is extended, the falls themselves may ultimately be completely drained.

We must now consider the question as to how the energy of the waterfall is converted into mechanical work at the shaft of our water turbine. It will be found that the vast majority of water turbines used at the present time can be classified in two groups: (i) The reaction type or "Francis" turbine; (ii) the impulse type or "Pelton" wheel; and it will be recalled that this classification is similar to that obtaining in steam turbine practice. We have already seen in that section of our work dealing with pumps,[2] that a "head" of water, i.e. a column of water of a certain height, exerted a pressure at the base of the column which was determined by the simple formula:—

$$p = w\,h$$

where p = pressure in lb. per sq. ft.;
w = weight per cu. ft. of water (62·3 lb. approx.);
and h = head in feet.

Hence, if we have a water turbine which is supplied through a closed pipe and the level of the free water is h ft. above the turbine, then it is obvious that the water will exert a pressure of p lb. per sq. ft. on the various parts of the turbine. If now we have a set of fixed guide vanes which deflect the water on to a series of vanes mounted on a shaft to form a wheel so that the water enters the wheel without shock, and these moving vanes are so shaped that the water is discharged from the wheel into the atmosphere with the minimum velocity, we shall have abstracted the energy from the water and in so doing have caused rotation of the shaft. The conditions will, in fact, be very similar to those in the reaction type of steam turbine,[3] except that, instead of having a number of stages as is usual in steam turbine practice when the reaction principle is employed, there is only one. Another way of considering the problem is to compare the water turbine to a centrifugal pump. In the latter, the wheel is rotated by means of some external source of power, and water at atmospheric pressure is drawn through the wheel and discharged through the guide vanes in the casing, where it acquires a considerable pressure enabling the water to be lifted to a considerable height above the pump. If now we supply the water to the pump from a certain height, thus obtaining a pressure in the pump chamber, the water would move in the reverse direction through the guide vanes and through the wheel, causing it to rotate. Thus the reaction type of turbine, or the Francis turbine,

[1] For the year ending October 31, 1922.
[2] See Chapter IV of this volume.
[3] See Volume I, Chapter IV, for an explanation of the principles underlying the operation of this type of steam turbine.

as it is generally called, can be regarded as a reversed centrifugal pump. The principal features of the design of a simple form of Francis turbine will be gathered from a study of Fig. 161, which shows a sectional elevation of the wheel and the guide vanes of a unit of 715 h.p. designed and constructed by Messrs. Escher Wyss et Cie, of Zurich, Switzerland, the famous firm of hydro-electric engineers to whom we are indebted for particulars and illustrations of a number of water power plants included in this chapter. The turbine is installed at Thun, in Switzerland, where the available head of water is only 6·1 metres (20 ft.). At normal load, the turbine has a speed of 150 revs. per min. and the quantity of water passing through the wheel in the same time is 11,000 litres (390 cu. ft.).

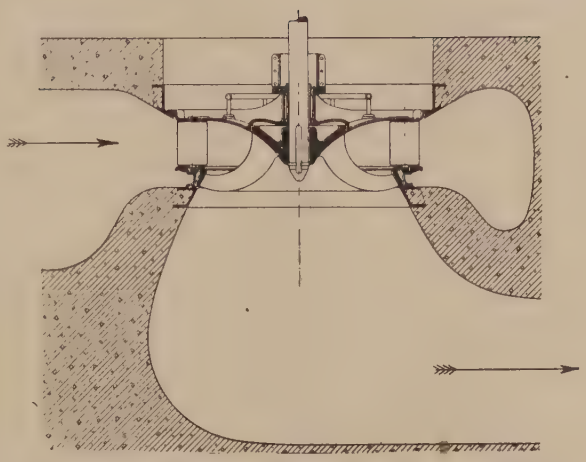

FIG. 161.—Sectional Elevation of Francis Turbine of 715 h.p. working under a head of 20 ft. Designed and constructed by Messrs. Escher Wyss et Cie, Zurich, Switzerland.

The principle outlined above forms the basis of a very large proportion of the water turbines which are employed at the present time in hydro-electric power stations. Indeed, except where the water is available at a very high head, thus yielding a considerable pressure at the power station, it will be found that the Francis type of turbine in some form or other is generally employed. As might be expected, the form which the turbine takes when the head is very small, say about 4 ft., is somewhat different from that when the head is, say, 500 ft.; but as we shall see as we proceed, the principle upon which the turbine works is substantially the same. While there is no hard-and-fast rule to guide the designer as to the maximum head at which the Francis type turbine can be employed, it will be found that for heads greater than 500 ft. the impulse type of turbine, or the Pelton wheel as it is commonly called, is generally adopted, while occasionally Pelton wheels will be found working at lower heads than this. At the time of writing the greatest head in the world which is employed for operating hydro-electric plant is to be found at Fully, in the Rhône Valley of Switzerland, where four Pelton wheels each developing 3,000 h.p. at a speed of 550 revs. per min. have been installed. These are supplied with water at a head of 1,650·7 metres, i.e. 5,420 ft., obtained from the hills above.

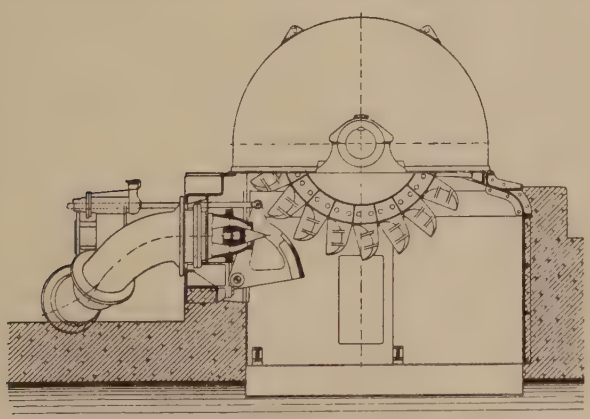

FIG. 162.—Elevation of Pelton Wheel of 8,250 h.p. working under a head of 1,120 ft. Designed and constructed by Messrs. Escher Wyss et Cie.

The principle governing the operation of the Pelton wheel, i.e. the impulse type of water turbine, is precisely the same as that employed in the impulse type of steam turbine as exemplified by the de Laval

turbine,[1] except that instead of a jet of steam we have a jet of water. Owing to the head of water above the turbine, it is, as we have just seen, at a very high pressure. If now this water is allowed to escape into the atmosphere through a suitably designed nozzle, it will issue forth at considerable velocity, and if, further, this jet of water is made to strike a series of blades, or buckets as they are termed, mounted on the rim of a wheel, it will cause the wheel to rotate. The general features of the design of a Pelton water-wheel will be understood from a study of Fig. 162, which shows in outline one of a number of units of 8,250 h.p. designed and constructed by Messrs. Escher Wyss et Cie, Ltd., for the Aluminium Company of Neuhausen, Switzerland, the plant being installed at Borgne, where the available head of water is 342 metres, i.e. 1,120 ft. At full load the discharge of water through the nozzle is 2,230 litres per sec. (78·9 cu. ft. per sec.), the speed of the wheel being 300 revs. per min. It will be noticed that the water, on issuing from the jet, strikes the buckets which are mounted on the periphery of the wheel, after which it falls into the pit below the wheel with practically no velocity, so that all the energy of the jet is absorbed by the wheel. Attention may also be drawn to the deflector plate mounted on the quadrant in front of the nozzle. This is connected with the governor and controls the speed of the wheel by deflecting a portion of the water of the jet from the buckets, the water which strikes the deflector plate falling directly into the pit without doing useful work. A good idea of the method adopted for supplying water at very high head to the turbines will be gathered from Fig. 163, which is a reproduction of a photograph showing the pipe lines supplying the hydro-electric power plant at Ritom, Switzerland. It will be noticed that the water is led from a lake or some other available source of supply high up in the hills to the power house in the valley,

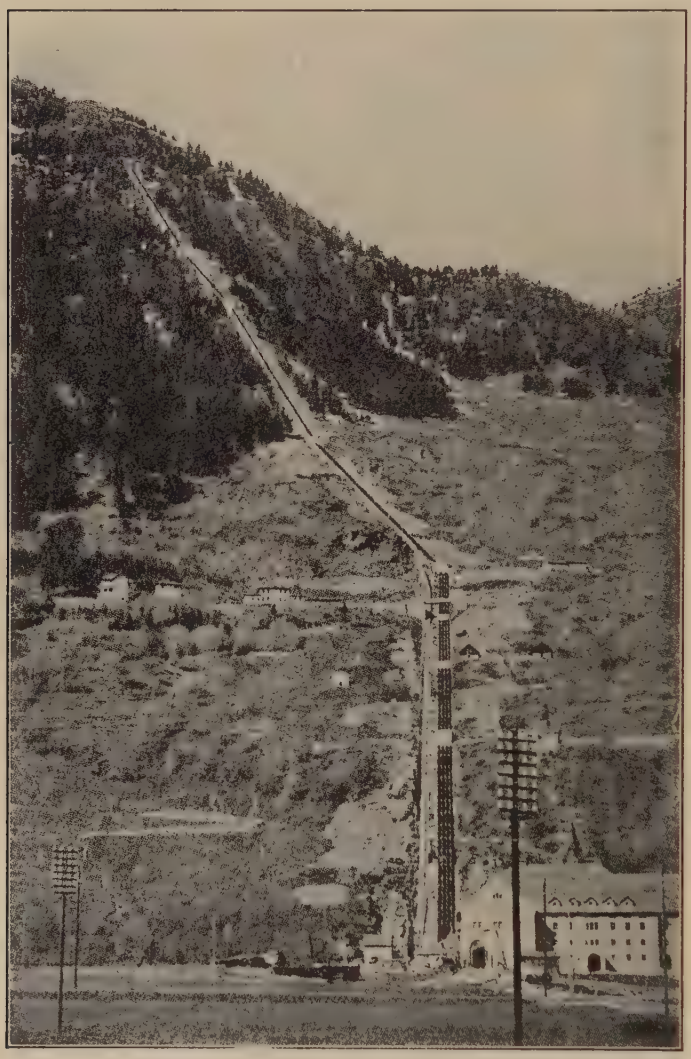

Fig. 163.—View showing Pipe Lines supplying water to the Hydro-electric Power Station at Ritom, Switzerland.

[1] See Volume I, Chapter IV.

where, after doing work in the turbines, it passes away to join a stream or river lower down. The exact velocity with which the water issues from the nozzles will, of course, depend upon the pressure which in turn depends upon the head. It is shown in text-books on elementary mechanics that, if any body falls through a height h feet under the influence of gravity, its velocity v expressed in ft. per sec. at the end of the fall is given by the expression:

$$v = \sqrt{2gh}$$

where g is the acceleration due to gravity, i.e. 32 ft. per sec. per sec. In the case of water falling through a pipe, the velocity is calculated from the same formula, but an allowance has to be made for the loss of velocity due to friction in the pipe. Thus in the case of the Fully installation to which reference has just been made, the velocity of the water, after falling through 5,420 ft., will be $\sqrt{2 \times 32 \times 5{,}420} = \sqrt{334{,}000} = 578$ ft. per sec., or taking friction into account (the loss of velocity due to this will depend upon many conditions such as nature of surface of pipe, size of pipe, etc.), the velocity of discharge may be taken as $578 \times 0{\cdot}93 = 538$ ft. per sec. An express train travelling at 60 miles per hour has a speed of 88 ft. per sec., so that the jet is moving more than six times as fast as an express train. This will give some idea of the speed of the jet as it strikes the buckets on the wheel.

One last problem remains to be considered: what is the relationship between the velocity of the water and the energy it contains, or, in other words, what is the amount of power which can be developed from such a jet. If a body of weight W lb. moves with a velocity v ft. per sec., then its kinetic energy is given by the expression:

$$\text{K.E.} = \tfrac{1}{2}\frac{Wv^2}{g} \text{ ft.-lb.}$$

Thus in every pound of water issuing from the jet, since W = 1, there is $\frac{v^2}{2g}$ ft.-lb. of energy. If it were possible to convert all this energy into work on the wheel, then for every pound of water which strikes the wheel, $\frac{v^2}{2g}$ ft.-lb. of work would be available at the shaft, neglecting losses due to friction at the buckets, at the shaft bearings, etc. In practice, the wheel is so designed that the water of the jet, after striking the buckets with high velocity, falls dead into the space below the wheel with practically no velocity at all, so that nearly all its kinetic energy is given up to the wheel. Hence if W lb. of water are discharged through the orifice per minute, neglecting losses, the work done on the wheel is

$$\frac{Wv^2}{2g} \text{ ft.-lb. per min., and}$$

$$\text{Horse power} = \frac{Wv^2}{2g \times 33{,}000}.$$

Or, taking η as the efficiency of the wheel, the horse power available at the shaft will be

$$\frac{\eta Wv^2}{2g \times 33{,}000},$$

while the capacity of the electric generator which such a wheel is capable of driving is

$$\frac{\eta Wv^2}{2g \times 33{,}000 \times 1{\cdot}34} \text{ kilowatts.}^1$$

We must now consider in greater detail the design of various practical types of water power turbines, after which a few particulars of actual installations can be given. Consider-

[1] N.B.—1 kilowatt = 1,000 watts, and 746 watts = 1 horse power, so that 1 kw. $= \frac{1{,}000}{746} = 1{\cdot}34$ h.p.

ing first of all the Francis or reaction type of turbine. When only a low head of water is available, an " open " construction is adopted—that is to say, the guide vanes do not draw their supply of water from a spiral casing, as in the case of a reversed centrifugal pump, but from the main body of water in the penstock.[1] The Francis wheel shown in Fig. 161 is of the open type. In this case, the turbine has a vertical shaft and the electric generator must be designed accordingly. It is not, however, essential that a vertical shaft should be employed, although for low heads this arrangement is obviously to be preferred, since every part of the wheel is then working under the same head. With a wheel of large diameter mounted on a horizontal shaft, the upper part of the wheel would be working under a different head from the lower part, a condition which would lead to a lower efficiency of the turbine.

The general layout of a Francis turbine with a horizontal shaft is shown in Fig. 164, which is also one of the designs worked out by Messrs. Escher Wyss. In this case the water discharges through a steel or iron draft tube into the tail race as shown. The main features of the design of one of these turbines will be gathered from a study of Fig. 165, which shows a single Francis turbine with horizontal shaft of somewhat similar construction to that shown in outline in Fig. 164. The ring carrying the guide vanes, and the tips of the vanes themselves, can be clearly seen, as also can the shaft which controls the position of these vanes relatively to the runner within. The function of this control gear is to regulate the speed of the turbine. The large cast-iron ring on the right is secured to the wall of the power house, the water being to the left of this ring. Hence access can always be obtained to the main shaft bearing, while the turbine runner can be withdrawn into the engine room when desired, access of the water to the penstock being first of all prevented. The flange taking the draft tube will also be noted. One great advantage of the horizontal shaft arrangement is that when units of large power are required, it is possible to mount two or more turbines on the same shaft, the water being led from a common penstock through the various wheels to a common tail race. An interesting example of a unit of this type is shown in Plate VII. This is a reproduction of a photograph of a sectional model prepared by Messrs. Escher Wyss, showing

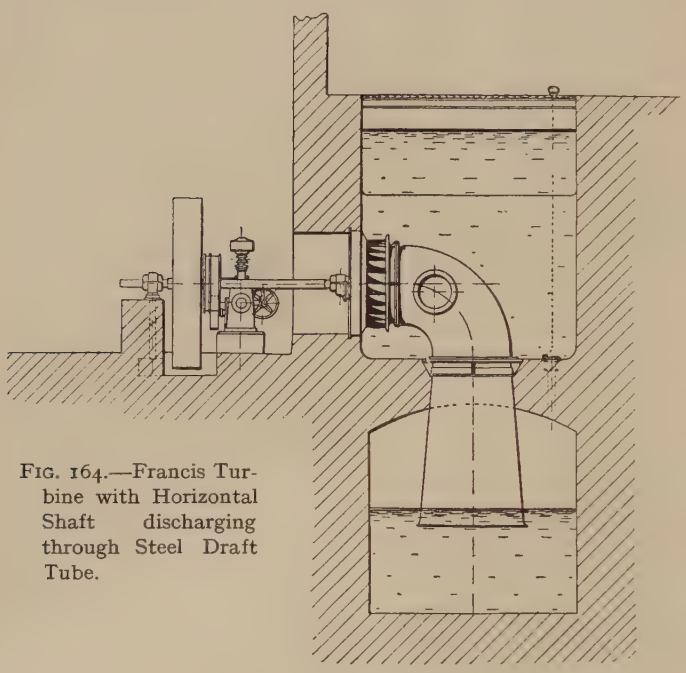

Fig. 164.—Francis Turbine with Horizontal Shaft discharging through Steel Draft Tube.

[1] The penstock can be regarded as that space in which the water is accumulated before it passes through the turbine whence it discharges into what is termed the " tail race." When high falls are being dealt with, the space into which the water is drawn before passing into the pipe line is termed the " head race." The terms " penstock " and " head race " appear to be used by some writers as though they mean one and the same thing.

the power station at Angst, on the River Rhine near Basel, Switzerland, which was equipped by them with ten horizontal quadruple turbines each developing 3,000 h.p. and direct coupled to an alternating current generator. The turbines work under a head of 8·75 metres, i.e. 28·7 ft. In the model the left-hand turbine is shown with the casing and the guide vanes removed, thus revealing the two wheels, but the right-hand turbine is shown as erected. The water enters through the screen shown on the extreme left of the picture and discharges through the concrete passages to the tail race. It may be noted that each unit really consists of two double runner turbines coupled together, the double runner unit being a type which is very frequently adopted in practice.

If the student has thoroughly understood the principles underlying the action of the centrifugal turbine pump, he should have but little difficulty in understanding the elementary theory of the Francis turbine. In both cases the guide vanes and the vanes on the wheel must be so designed that the water passes through the various parts of the turbine without shock; but whereas in the pump the chief object was to discharge the water from the impeller with maximum velocity, the velocity energy being subsequently converted into pressure energy in the guide vanes, in the Francis turbine the water at the pressure corresponding to the head under which the turbine works passes through the guide vanes, where its pressure energy is converted into velocity energy, whence it passes through the wheel the vanes on which are so designed that it leaves the wheel and passes to the tail race with minimum velocity. The change of velocity which occurs as the water passes through the turbine wheel can be easily determined by drawing the parallelograms of velocities for the water as it enters and leaves the wheel. A diagrammatic section of a Francis turbine is shown in Fig. 166. It will be noticed that the guide vanes arranged outside the wheel form converging nozzles through which the water passes with increasing velocity, so that when it reaches the tip of the guide vane its velocity V_1 is considerable, the velocity being in a direction tangential to the guide vanes as shown. The wheel is rotating so that the tips of the vanes on the wheel are moving with a velocity tangential to the outer rim of the wheel. Let this velocity be V_2. Then by the usual parallelogram rule, the velocity of the water relatively to the vanes on the wheel is V_3, and if this velocity is tangential to the vanes, the water will enter the wheel without shock.

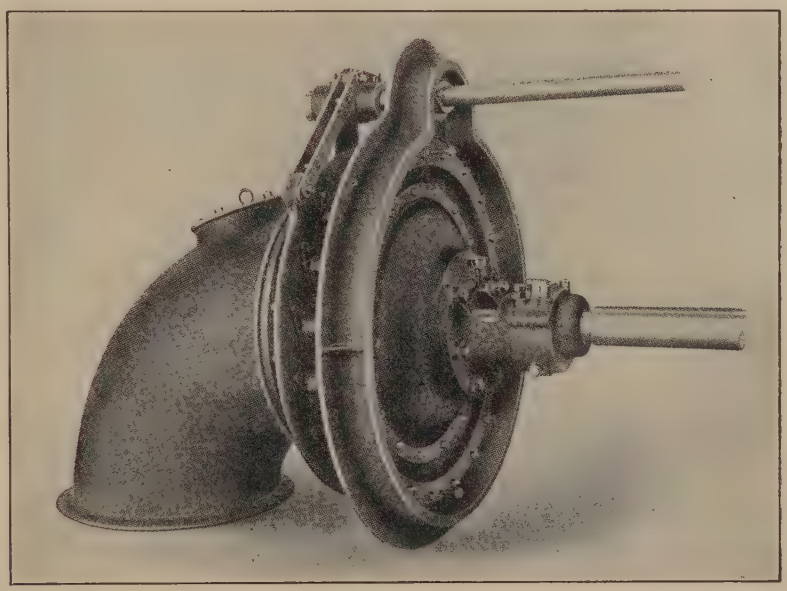

Fig. 165.—Francis Turbine with Horizontal Shaft, showing Guide Blades and Control Gear.

Another way of considering this parallelogram is to assume that the vanes on the wheel have a definite angle and it is desired to find the speed of rotation of the wheel.

As before, if V_1 is the magnitude and direction of the velocity of the water leaving the guide vanes and this water enters the wheel without shock, i.e. in the direction of V_3 tangential to the vanes on the wheel, then without knowing the magnitude of V_3, since the direction of V_2 is also known, this being tangential to the wheel, the parallelogram can be completed, giving the magnitude of V_2, i.e. the speed of rotation of the wheel. In other words, V_2 is the speed at which the turbine will run under conditions of maximum efficiency since at slower or faster speeds the water will enter the wheel with shock, causing loss of energy. It will be seen that the water enters the wheel with a velocity V_3 tangential to the vanes, and, neglecting friction, it can be assumed that it passes through the wheel at practically constant velocity, although actually there must be a slight loss of velocity due to friction. Moreover, if the wheel is incorrectly designed there will be a restriction of the area of the passage between the vanes on the wheel, as shown in the diagram, thus causing the velocity of the water to increase as it passes through the wheel. Actually, however, the wheel is so designed that the area of the discharge passage is as large as possible, so that the velocity of the water leaving the wheel is as small as possible. (This increase in the area of the outlet passage is clearly shown in Fig. 179, page 205, which shows a section through a Francis turbine of 10,000 h.p.) Returning now to our diagram, we can assume that v_3, the velocity of the water relatively to the wheel as it leaves the vanes, is equal to V_3, and that its direction is tangential to the vanes at the point of exit. But the wheel itself is rotating so that the tip of the vane has a velocity v_2 tangential to the inner circumference of the wheel. Hence the resultant velocity of the water leaving the wheel is v_1. It will be noticed that this is very small and that its direction is practically towards the centre of the wheel. Thus the water leaves the turbine with the minimum velocity, or, in other words, the maximum proportion of its energy is imparted to the wheel.

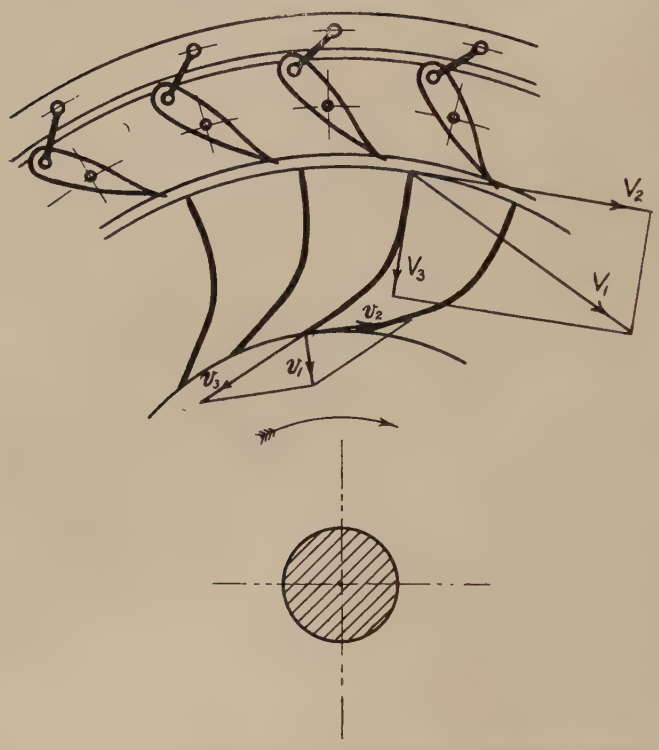

FIG. 166.—Diagrammatic Section of Francis Turbine, showing Change of Velocity of the Water passing through the Wheel.

Attention has already been drawn to the fact that in the Francis turbine the guide vanes are movable. It will be noticed from the diagram that each vane works on a pivot and that it is secured by a short link to an outer ring. If now this outer ring is turned about its axis, it will cause the guide vanes to turn about their pivots, thus altering the shape of the space between the vanes. In this way not only is the velocity V_1 changed, but the amount of water which enters the wheel can be varied. This offers a very simple method of governing the speed of the turbine, and it will be found that in practically all designs of Francis turbine, the speed is automatically controlled by varying the position

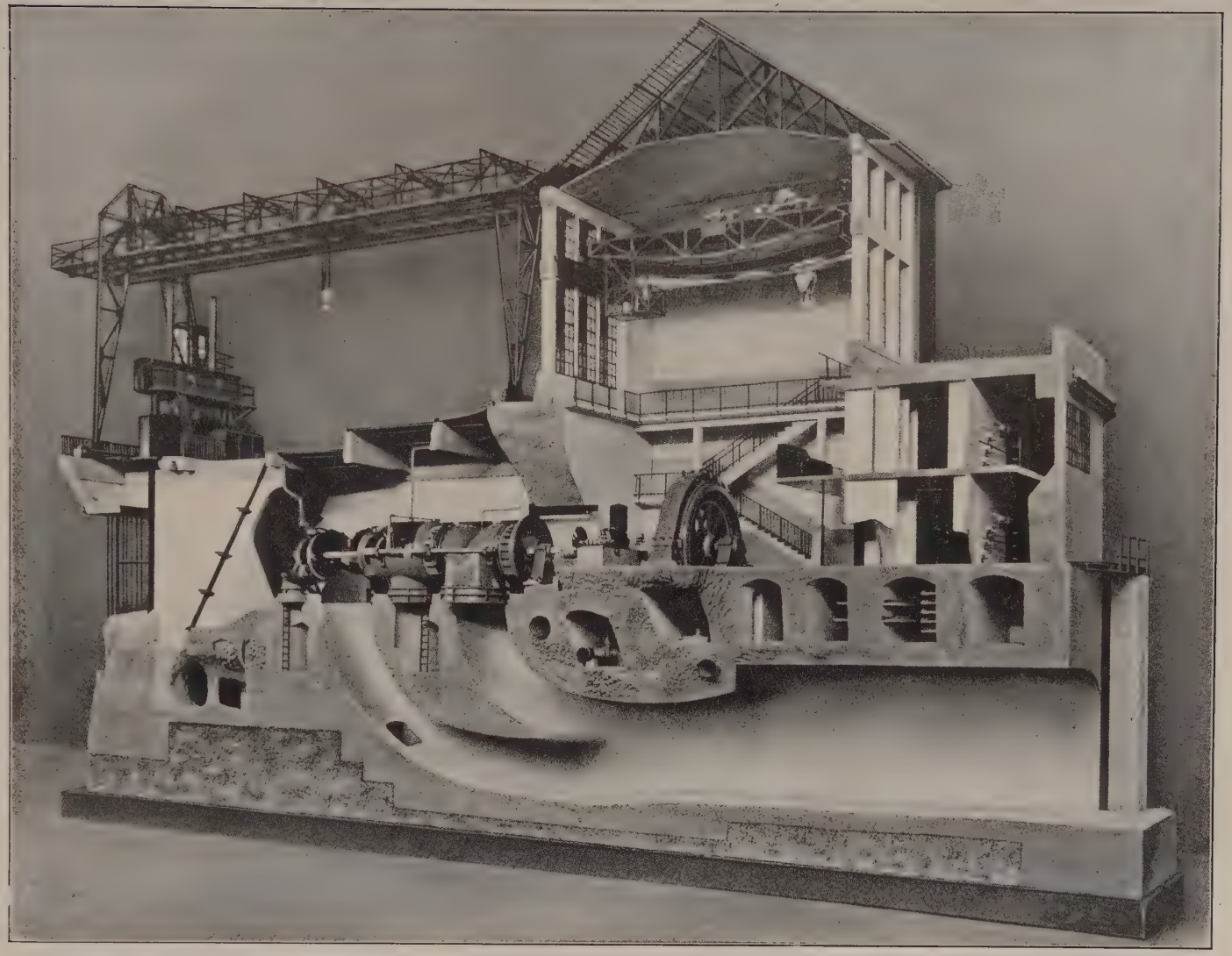

PLATE VII.—SECTIONAL MODEL OF HYDRO-ELECTRIC POWER STATION AT ANGST ON THE RIVER RHINE, NEAR BASEL, SWITZERLAND, SHOWING ONE OF THE TEN HORIZONTAL-SHAFT QUADRUPLE FRANCIS TYPE TURBINES, EACH DEVELOPING 3,000 H.P. UNDER A HEAD OF 28 FT. THIS STATION WAS DESIGNED AND EQUIPPED BY MESSRS. ESCHER, WYSS ET CIE, ZURICH.

of the guide vanes in accordance with the load on the electric generator which is coupled to the turbine. The reader can study again Fig. 165, and he will realise that by a slight turning movement of the small shaft which is shown above the main turbine shaft, the ring carrying the links is given a slight movement of rotation, and this in turn moves the guide vanes. The movement of the small shaft is controlled by the governor, and thus a uniform speed of rotation can be maintained.

We are now in a position to understand the details of two or three typical installations of Francis turbines. It has already been noted that these turbines can be designed to work at very low heads as well as at heads exceeding 500 ft., and as an example of an installation in which only a low head is available we will consider the plant laid down for the City of Chester, which is of interest as being the first hydro-electric plant laid down in

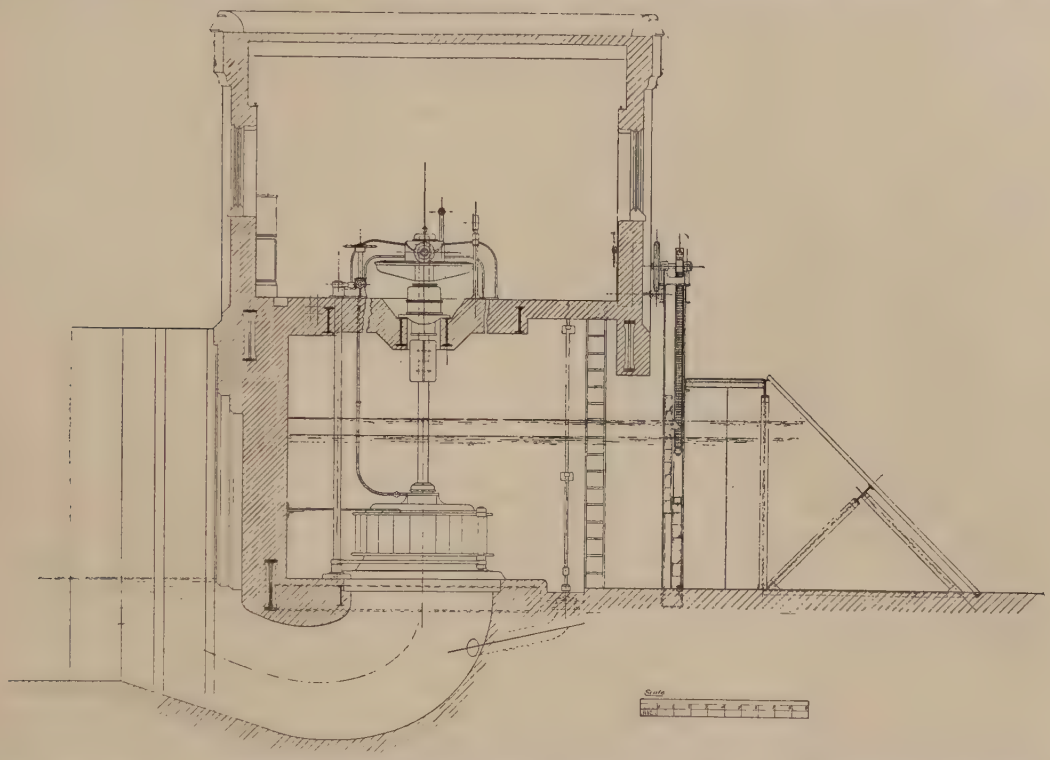

Fig. 167.—Section through the Chester Hydro-electric Power Station, showing Francis Turbines, with Forebay and Tail Race.

this country in connection with a public electricity supply undertaking. Reference has already been made to this installation in the introductory chapters of this work,[1] but a more detailed description of the plant may now be given. In designing the plant, special consideration had to be given to the fact that the river at Chester is affected by tidal waters, and it was necessary to adopt specially designed turbines and dynamos capable of working efficiently over a wide range of speeds. It was determined that the average amount of fresh water available at different months of the year is: December, 300,000 cu. ft. per min.; January, February, and March, 100,000 cu. ft. per min.; April, May, October, and November, 50,000 cu. ft. per min.; and July, August, and September, 20,000 cu. ft. per min., the lowest dry weather flow being about 10,000 cu. ft. per min. The effect of this varying flow is to give the following average head: December, 4·5 ft.; January, February, and

[1] See Volume I, Chapter II, where a photograph of the interior of the power house will be found.

March, 6·25 ft.; April, May, October, and November, 7·5 ft.; June, July, August, and September, 8·75 ft. In order to make use of this flow, a weir was placed diagonally across the river at the east side of the Old Dee Bridge, thus forming a channel to conduct the water to the power house, the water passing through the necessary strainer rack and sluice gates before reaching the building. A cross-sectional elevation of the building showing the forebay and the tail race is reproduced in Fig. 167, while an interesting view showing the interior of the building during construction is shown in Fig. 168. Particular attention may be drawn to the tail race from each turbine, these being made of concrete moulded to the correct form. It will be noticed that three turbines are installed, these being of the vertical shaft Francis type, the general form of the turbines, together with their generators, being shown in the sectional drawing.

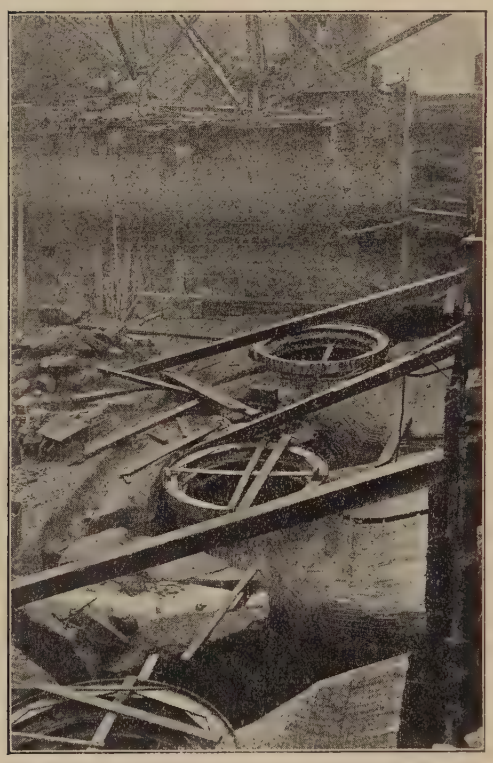

Fig. 168.—Interior of Chester Hydro-electric Power Station. Tail Race from the Turbines is under construction.

The turbines were specially designed by Messrs. James Gordon & Co. for working under a head of water varying from 1 to 9 ft., two of them being designed for dealing with 30,000 cu. ft. of water per min. under the maximum head of 9 ft. and developing 415 b.h.p. at a speed of 50 revs. per min., while the third is designed for dealing with 22,000 cu. ft. per min. under similar conditions and is capable of developing 305 b.h.p. at a speed of 55 revs. per min. The turbine runners are constructed with vanes of forged steel plate securely cast with the centre boss and the surrounding ring, which are of cast iron. The guide blades are also of cast iron bushed with gunmetal and mounted on steel spindles. The vanes are coupled by links or shafting to the hand regulating gear, which is so designed that the vanes, which have ground edges, can be brought together, thus ensuring a water-tight pit. This gear is mounted on the generator floor and is provided with an indicator showing the position of the guide vanes. The weight of the turbine wheel, together with the shaft and the bevel gear wheel through which the generator is driven, is supported by a collar suspension bearing fitted at the upper end of the vertical shaft on the generator floor as shown in Fig. 167, the thrust rings of this bearing being submerged in oil. Before entering the turbines, the water passes through a wrought-iron strainer rack fitted across the full width of the head race, while sluice gates are provided to each turbine, these being operated from inside the power house.

The turbines are connected through machine-cut steel helical bevel gears and flexible couplings to the electric generators, which are of the shunt wound interpole type, constructed by the Lancashire Dynamo & Motor Co. The two large generators have an output of 225 kw. at a speed varying between 143 and 285 revs. per min., while the output of the smaller generator is 185 kw. at any speed between 167 and 300 revs. per min. Although the normal speed of each generator is in the region of 300 revs. per min., the armatures are specially constructed so that they may be safely run at a speed of 500 revs. per min.

without fear of damage, as might occur when the load is suddenly thrown off the machine ; while in view of the fact that the machines are compelled to work in a damp atmosphere special attention has been paid to the insulation throughout. The current is generated at a pressure of 440 to 500 volts, and after passing the switchboard arranged at one end of the power house, it is transmitted in lead-sheathed steel-armoured cables laid direct in the ground to a sub-station in the town, from whence it is distributed to the consumers. As the hydro-electric station generates more than sufficient energy to meet the requirements of the district supplied by this sub-station, the balance is sent to the steam-operated generating station, where it passes through a reversible booster and is either sent out to other areas of supply or stored in a battery in readiness for the next peak load.

Some idea of the economy which is effected by the use of water power for generating electrical energy can be gathered from the figures given in a report made by Mr. S. E.

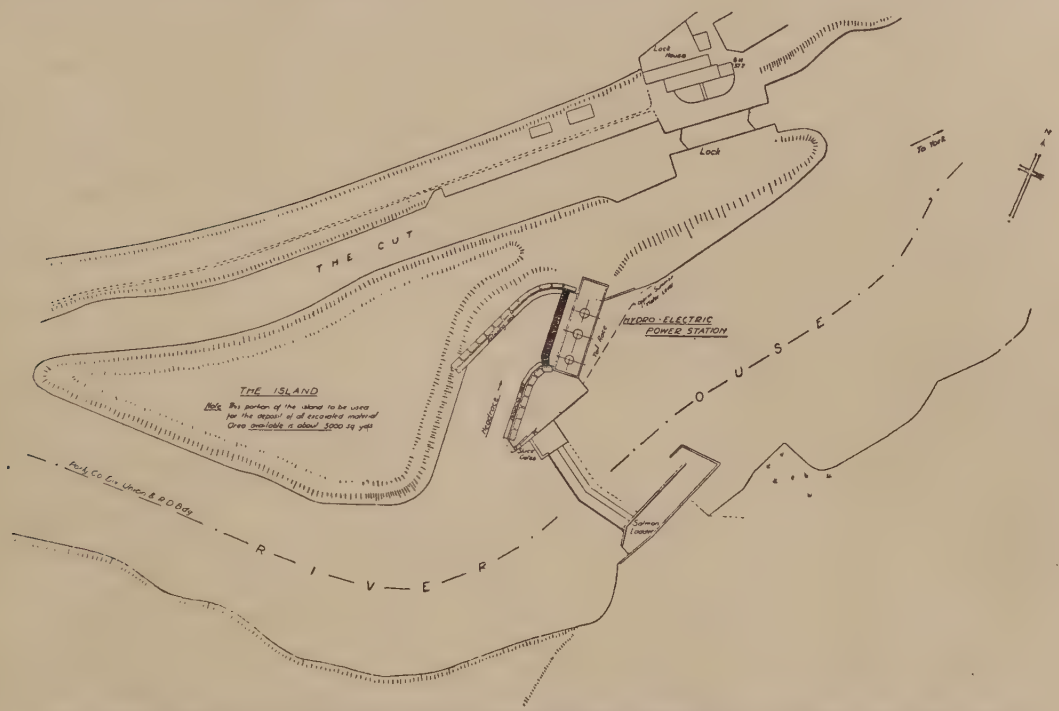

FIG. 169.—Site of the Linton Lock Hydro-electric Power Station on the River Ouse near York, showing Weir, Head Race, and Tail Race.

Britton, M.I.E.E., M.I.Mech.E., the City Electrical Engineer at Chester. The cost of generating electricity from the water power plant works out at 0·2757*d.* per unit as compared with 1·173*d.* per unit when generated from steam power. Considering the expenditure and the receipts over a period of five years, the report states that the total income from the sale of electricity was £116,152, of which £56,540 was derived from current generated by the steam installation and £59,612 from the hydro-electric installation, the average receipts from both stations being 1·96*d.* per unit. After deducting the necessary revenue and capital charges incurred by the two generating stations, a balance of £14,368 remains from the sale of current produced by steam and £49,002 from current produced by water power, together providing a sum of £63,370 to meet revenue expenditure and capital charges on accumulators, distribution, and administration, totalling in all £47,820, which leaves a surplus of £15,500, of which it will be recognised that the hydro-electric power station is responsible for a very big proportion. Apart from the monetary saving, it is

estimated that this station results in a saving of 3,000 to 4,000 tons of coal per annum. This plant has now been in operation for more than ten years and has given every satisfaction, giving a clear indication of what can be done to utilise water power even when the available head is very small.

Another very interesting example of a hydro-electric power station in this country is to be found on the River Ouse at Linton Lock, about eleven and a half miles from York, where a fine water power plant capable of yielding a maximum of 1,190 b.h.p. has been laid down by the Electricity Committee of the Corporation of the City of York.

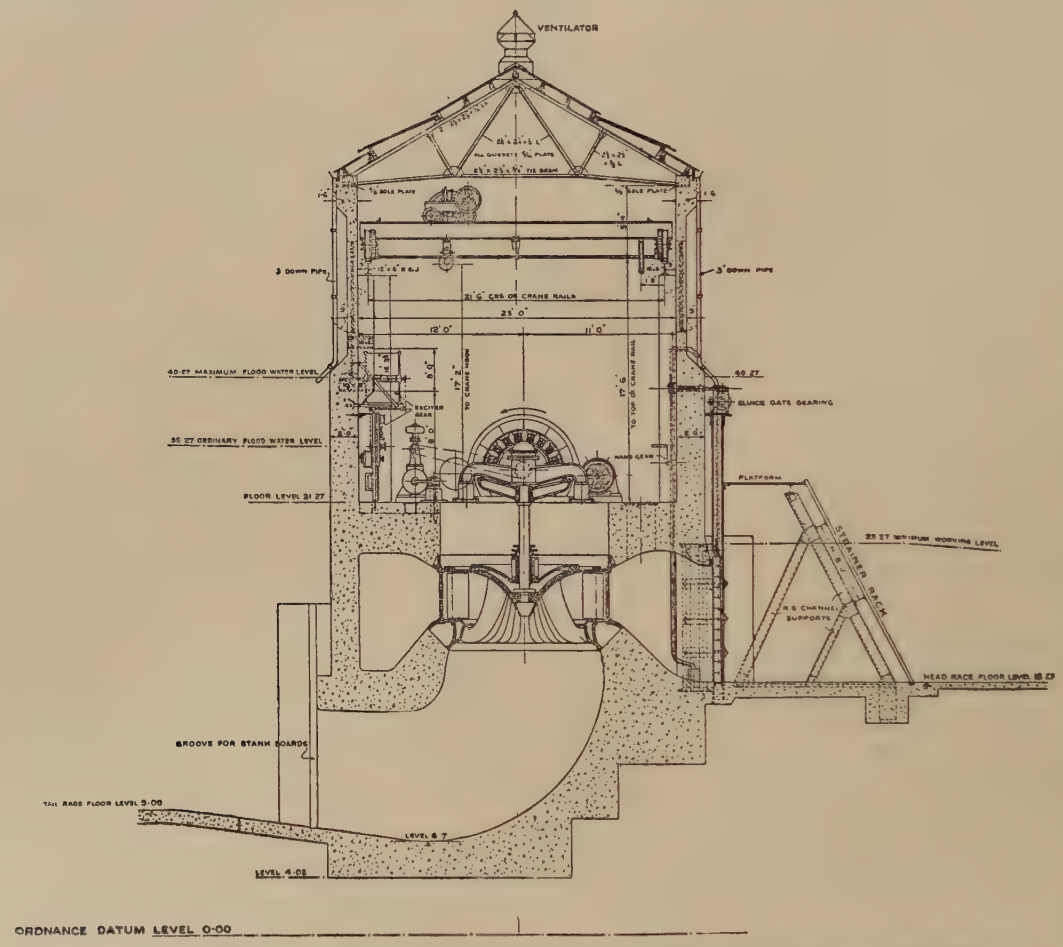

FIG. 170.—Cross Sectional Elevation of Power House at Linton Lock, showing one of the Francis Turbines. The station was designed and equipped by The English Electric Co., Ltd.

The plant consists of three turbines of the Francis type, two of 430 h.p. each and one of 330 h.p., all working under an average head of 8·83 ft., the two larger units being coupled together to drive through gearing a single generator of 540 kw. capacity, while the smaller turbine drives a generator of 210 kw. capacity. The contractors for the complete installation were the English Electric Co., Ltd., to whom we are indebted for the following illustrations and particulars of the plant. The general layout of the site at Linton Lock will be understood from a study of Fig. 169, which shows the River Ouse in the vicinity of the power house, it being recognised that the bend in the river would facilitate the arrangement of the weir and of the power station building. The River Ouse, like most rivers

without natural compensation, has a very variable discharge. Prior to going ahead with the scheme, it was determined that, roughly speaking, the discharge varied from 50,000 cu. ft. per min. in dry periods to 200,000 cu. ft. per min. in the wet months, while the available water level varied from 1 ft. 4 in. to 11 ft. 4 in. Taking 100,000 cu. ft. per min. as the average flow with an average head of 8·83 ft., it was found that about 1,200 h.p. was available, and a close calculation showed that it should be possible to obtain 2,750,000 kw.-hours of electrical energy per annum. In view of these figures, it was considered that the scheme was a sound one. As will be seen from the plan of the site, the water is led into the head race by means of a weir placed across the river, it being agreed with the owners of land on the riverside that the rise of water due to this weir

FIG. 171.—The Power House at Linton Lock, looking along the Forebay.

should not exceed 2 ft. 9 in. above that previously existing, while special precautions in the shape of retaining walls, stone paving, etc., were taken to protect the river bank against floods. The water on the left of the weir goes into the head race—or forebay, as it is sometimes termed—passes through a strainer rack made of wrought-iron bars 3 in. by $\frac{3}{8}$ in. in section placed 2 in. apart, whence it flows through the sluice valves and enters the power house. The relative position of the strainer rack and the sluice valves will be understood from a glance at Fig. 170, which shows a cross sectional elevation of the power house together with the head race and the tail race. Attention may be drawn to the form of the passages leading to and from the turbines. The power house itself is shown in Fig. 171, which is a reproduction of a photograph taken looking down the forebay. It may be mentioned that, owing to the silty nature of the soil, it was necessary to drive no fewer than 110 piles in order to ensure safe foundations for this building.

The choice of the number and of the size of the turbines was made with a view to obtain the highest efficiency under all conditions of load. It is generally known that the efficiency of water turbines working under low heads drops considerably when they are used at a load 50 per cent. below their rated output, so that in order to obtain the best efficiency, it is necessary to run them as far as possible always between half and full load. Consequently, when only small quantities of water are available, it is most economical to use turbines of correspondingly small full-load output. On the other hand, the larger the turbine, the better its efficiency and the lower the cost per unit of electricity generated. In the case of the Linton Lock installation it was found that the most economical arrangement was to employ three turbines of the size given above. By using these in accordance with the following table, it is possible to have the plant which is actually running always

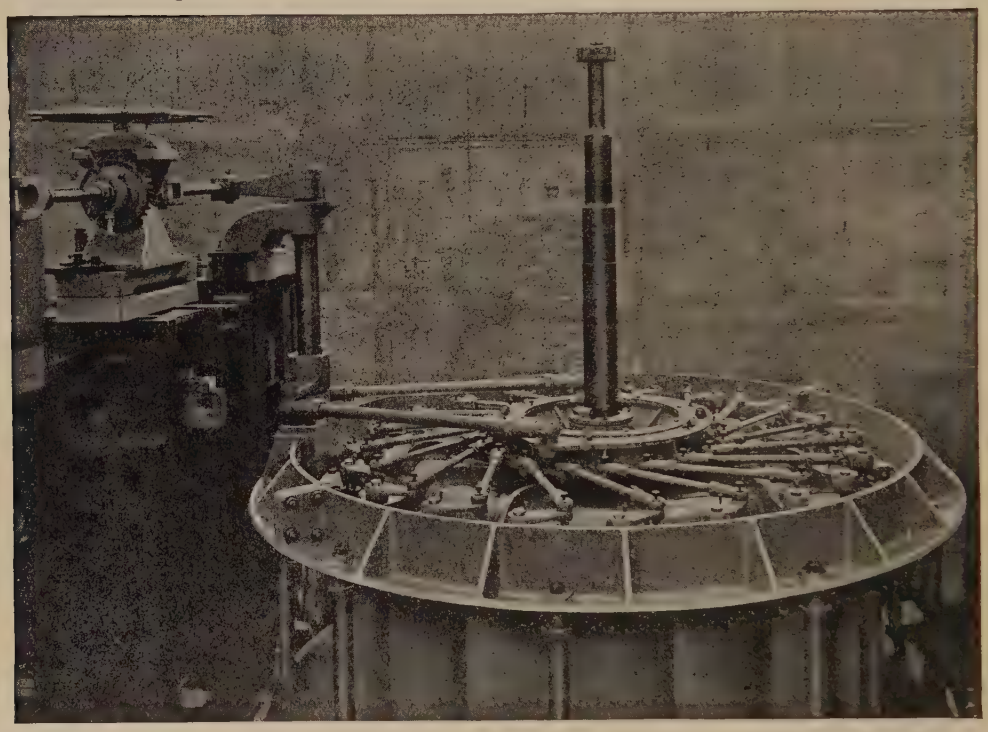

FIG. 172.—One of the Francis Turbines of 430 h.p. for the Linton Lock Hydro-electric Power Station, erected in the works of the makers, The English Electric Co., Ltd.

loaded up to more than 72 per cent. of its rated full load, apart from the lower limit in Case 3 or during a few exceptionally dry periods.

SCHEDULE FOR RUNNING OF WATER TURBINES ACCORDING TO LOAD ON STATION

No.	Plant Running.	Available Output.	Per Cent. of Full Load on Turbines.
1	Small Turbine	0–330 h.p.	0–100
2	One Large Turbine	330–430 h.p.	77–100
3	Small and One Large Turbine	430–760 h.p.	57–100
4	Two Large Turbines	760–860 h.p.	88–100
5	All Three Turbines	860–1,190 h.p.	72–100

This means that the turbines at the power station, on an average, work with an efficiency of 75 to 83 per cent. according to the available head.

The general form of the turbines is shown in the drawing of the power house (Fig. 170). It will be noticed that the guide vanes are mounted on a ring secured in the concrete spiral chamber, after passing through which the water is directed at the proper angle on to the runners of the wheel mounted concentrically inside the ring of guide vanes. Each turbine has twenty guide vanes, these being 43·3 in. high for the 430 h.p. turbines and 35·5 in. high for the 330 h.p. turbines. The guide vanes are made of cast iron and are pivoted in bronze bushes carried by the foundation rings. The foundation rings are also of cast iron, connected together by means of steel pillars and built into the concrete as shown on the drawing, thus forming a thoroughly rigid job. The guide vanes are operated by means of a system of levers and linkwork as shown in Fig. 172, which is a view of the turbine as erected at the Willans Works, Rugby, of the English Electric

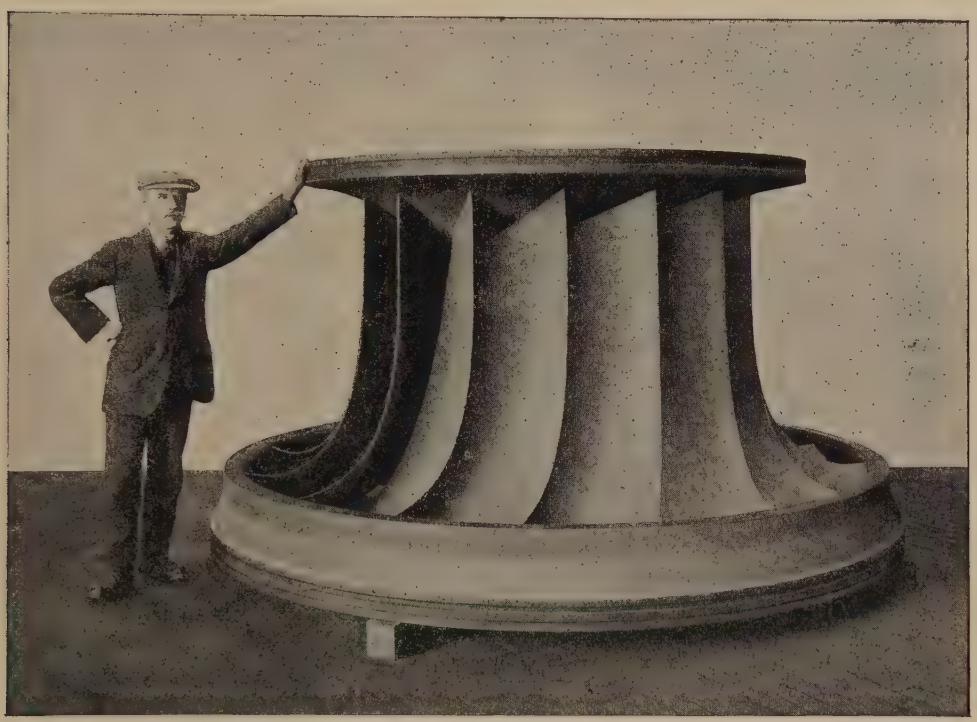

FIG. 173.—The Wheel of a 430 h.p. Francis Turbine for the Linton Lock Hydro-electric Power Station. The Average Head available is 8·83 ft.

Co. It will be noticed that a circular motion of the small vertical shaft to the left of the turbine causes the central ring to move on its axis, thus giving an angular motion to each of the guide vanes and so regulating the amount of water passing through to the wheel. The form of the upper foundation ring will be noted, as also will the position of the guide vanes below.

The turbine wheels were carefully designed in order to reduce the axial vertical thrust to a minimum. A view of the wheel for one of the larger units is shown in Fig. 173, and as will be noticed from this and from the sectional elevation of the wheel in the drawing (Fig. 170), the top cover of the wheel is extended beyond the vanes, while balancing holes are provided in this cover plate so that the space immediately above the wheel and beneath the cover of the turbine casing is under suction, and thus the effect of the extended cover of the wheel is to balance the pressure which the water exerts at the entrance of the vanes. For the large turbines, the wheel has an exit diameter of 10 ft.,

while its weight exceeds $5\frac{1}{2}$ tons, this being the largest wheel of its type installed in this country at the time of writing. The larger turbines have a normal speed of 60 revs. per min., and the small one a speed of 75 revs. per min., these speeds being controlled by means of automatic oil pressure governors. The servo-motor of the main generator controls the position of the vanes of both the large turbines, and if one of them is not working, the control apparatus is disconnected from the servo-motor ; at the same time the turbine itself is also disconnected from the generator. Oil pressure governors are employed on account of their reliability, the oil at the same time serving to lubricate the various bearings.

As already noted, the two large turbines are arranged to drive a single large generator, while a small generator is driven from the small turbine. In each case the generators

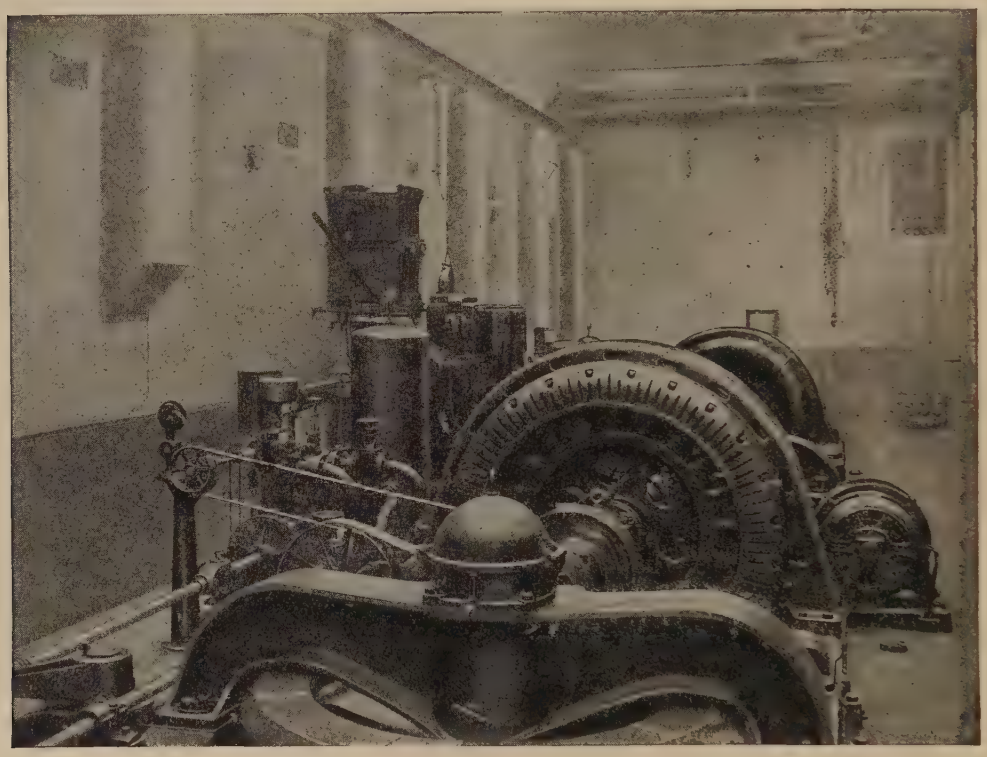

Fig. 174.—The Interior of the Hydro-electric Power Station at Linton Lock, showing the Generators and Bevel Gearing.

have horizontal shafts and are driven through double helical machine-cut bevel gearing and provided with a flexible coupling on the horizontal shaft. The generators are of the three-phase alternating current type working at a terminal pressure of 6,600 to 7,500 volts, the current having a frequency of 50 cycles per sec. The large alternator has a capacity of 540 kw. and runs at a normal speed of 300 revs. per min., while the smaller alternator of 210 kw. capacity has a normal speed of 375 revs. per min. Both units are, however, designed to withstand a runaway speed of about 1·9 times the normal speed. The full load efficiency of the generators is about 94 per cent., so that, taking 80 per cent. as the average efficiency of the turbines and allowing 2 per cent. for losses in the bevel gears, the overall efficiency of the station is about 74 per cent. The general appearance of the generators and of the gears will be gathered from Fig. 174, which is a view of the interior of the power house, the large generator which is situated between

the two large turbines being in the foreground, with the smaller generator in the background. The small machine immediately to the right of the large generator is the direct current exciting dynamo for this set. The heavy cast-iron beam mounted over the gearing

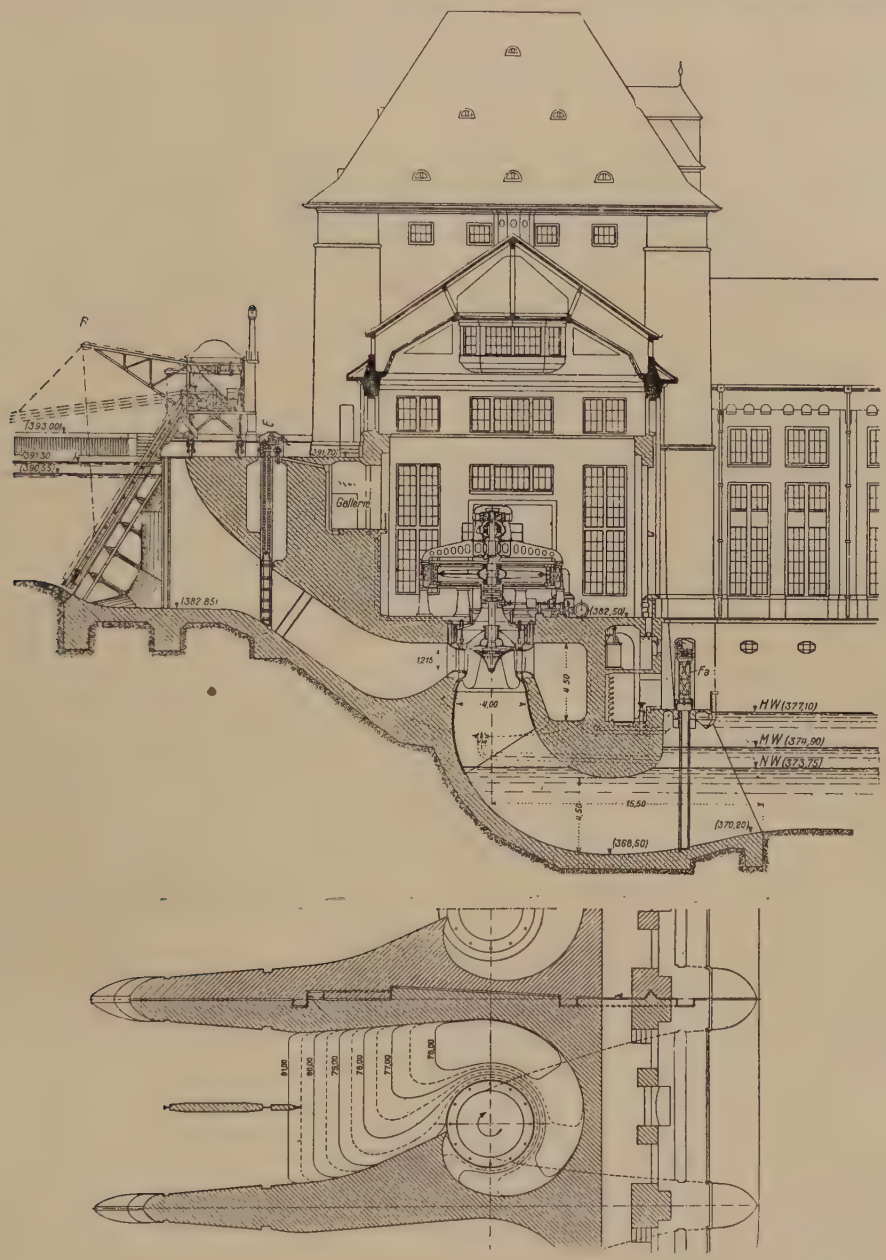

FIG. 175.—The Hydro-electric Power Station at Gœsgen, Switzerland, showing one of the 10,000 h.p. Francis Turbines, working under an Average Head of 53·4 ft. The plan shows the concrete spiral leading to the turbine.

carries the step bearing for the vertical turbine shaft. The switchboard for controlling the current from the generators to the feeders will be noticed on the left, the switch gear being enclosed in oil, as is usual when high voltage alternating currents are being dealt

with. The electric power is then transmitted to York, eleven and a half miles distant, through underground cables, a supply being given to the villages, etc., along the route. The installation at Linton Lock can be regarded as a fine example of a hydro-electric power house of moderate capacity, and it serves to indicate what can be done even in this country, where, generally speaking, only low heads of water are available.

It must not be imagined, as the foregoing descriptions appear to suggest, that where only low heads are available the power that can be obtained is necessarily small. The power developed by a water turbine depends not only upon the head, but also upon the volume of water available; the smaller the head, the greater the volume of water required per horse power. Provided, however, that sufficient volume of water is available, there is no reason, despite the existence of only a low head, why units of large power should not be employed. As an example of a low-head hydro-electric installation where units of large power are employed, we may briefly consider the power station at Gœsgen, in Switzerland, of the Olten-Aarbourg Electricity Supply Co. This station contains six Francis turbines each of 10,000 h.p. working under a mean head of 16·25 metres (53·4 ft.). The turbines, which were designed and constructed by Messrs. Escher Wyss et Cie, of Zurich, and are of the vertical shaft type, are direct coupled to alternators of 7,050 kw. capacity which generate current at a pressure of 7,700 to 8,400 volts at a frequency of 50 cycles per sec. The general layout of these units will be understood from a study of Fig. 175, which shows a sectional elevation through the power station taken at the centre of one of the units, together with a plan showing the concrete spiral through which the water is led to the turbine guide vanes from the head race. Attention may be drawn to the strainer racks and to the sluice valves which respectively clean and control the supply of water to the turbine. The various dimensions given on this drawing are in metres. A view of the wheel of one of the turbines taken as it is about to be lowered into position during the erection of the turbine in the power house is given in Fig. 176. Some idea of

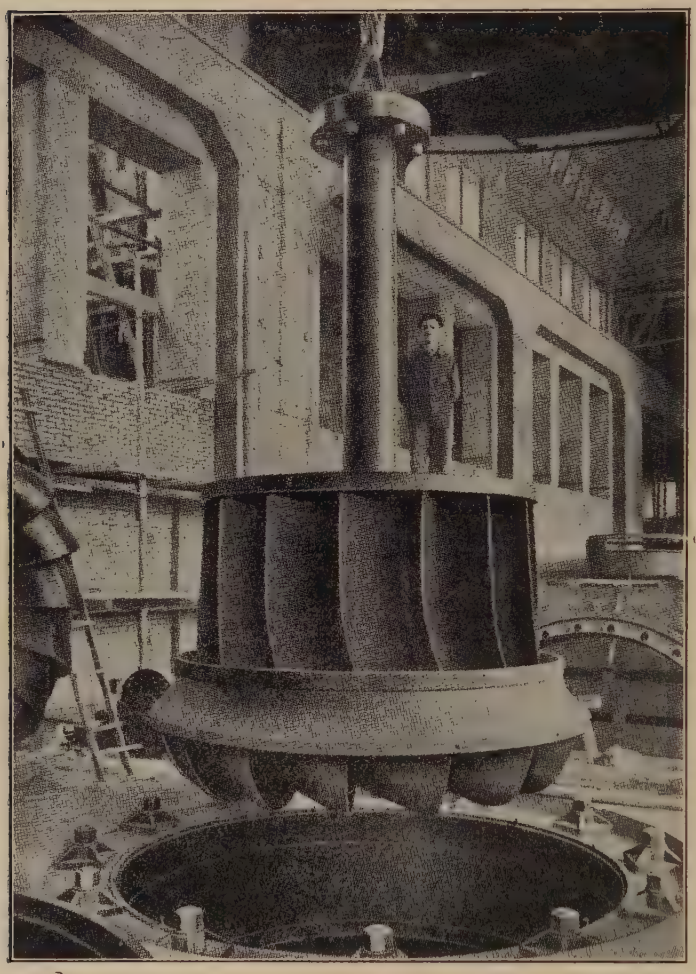

Fig. 176.—The Wheel of one of the 10,000 h.p. Francis Turbines for the Gœsgen Power Station being lowered into position.

the size of the wheel may be gathered by comparison with the size of the man standing on the top of the wheel. Actually, the wheel is 4 metres diam. (13 ft. 1½ in.) at its largest part, and it is designed at normal load to pass 300 cubic metres (10,600 cu. ft.) of water per sec., the speed of rotation being 83·3 revs. per min. A view of the interior of the power house showing the six main generators, which are of the vertical shaft type direct coupled to the turbine wheel, is reproduced in Fig. 177, and it is of interest to note the exciting dynamos which are mounted on an extension of the shaft right at the top of the unit.

When water turbines of the Francis type are being employed under the highest heads

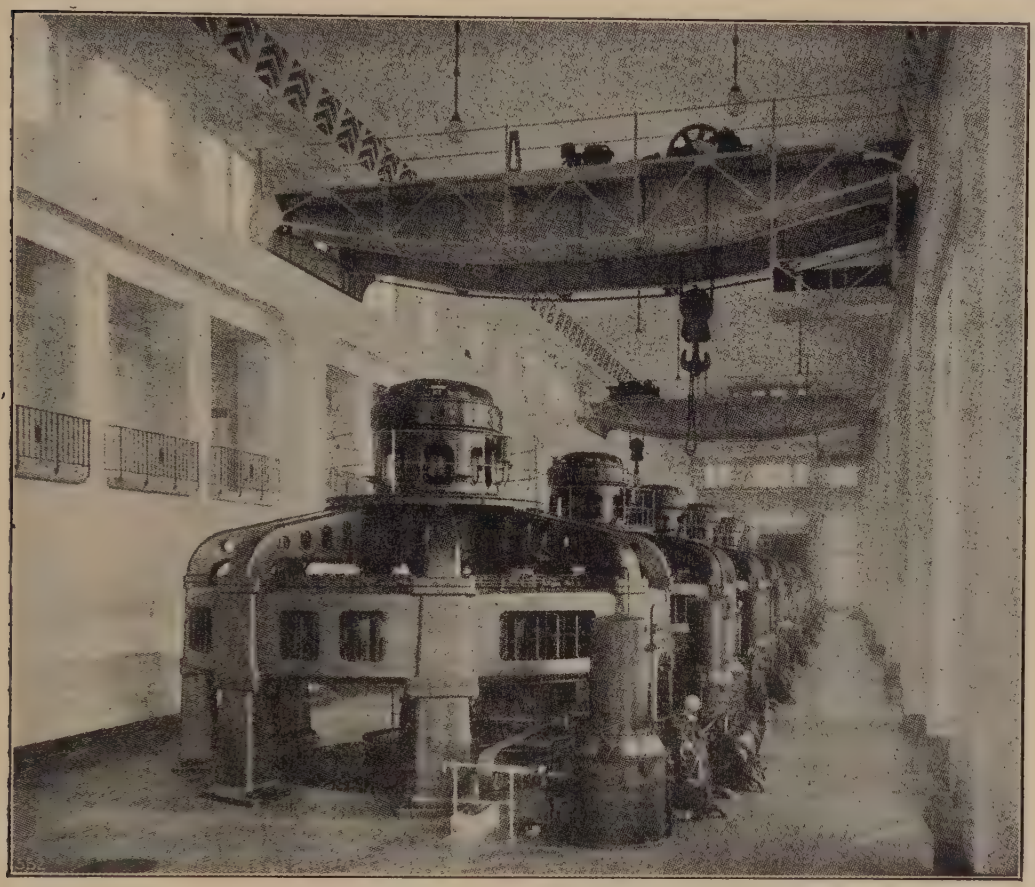

FIG. 177.—The Interior of the Power House at Gœsgen, Switzerland, showing the six 7,050 kw. Generators. Each is driven by an Escher Wyss Francis Type Turbine of 10,000 h.p.

for which this type is considered suitable, i.e. heads approaching 500 ft., instead of the more or less open construction which is characteristic of the units already described, it will be found that, in view of the smaller volume of water required in comparison to the size of the wheel, the turbine is provided with a cast-iron or steel spiral casing through which the water is led to the guide vanes, whence it passes to the wheel. At the same time, the blades on the wheel become of smaller relative area in view of the greater velocity of the water passing through the wheel. The general features of the Francis turbine having a cast-iron spiral casing are shown in Fig. 178. In this unit the water enters the spiral casing from above, passes round the casing, which becomes of progressively smaller and smaller cross sectional area, entering the guide vanes arranged around the turbine

wheel as shown. These guide vanes are adjustable in position in exactly the same way as already described for turbines of low head, and thus the speed of the turbine can be regulated as required. It frequently happens with turbines of this type that pipe lines are employed for leading the water from the head race to the turbine itself, while, as in the case shown, a pipe is employed to lead the water from the turbine to the tail race; but as will be at once realised, there are an almost infinite number of ways in which the water can be led to and from the turbine. The action of the water in the turbine itself is, however, precisely the same as in the case of the low head Francis turbines already described, the only difference being in the details of design due to the greater pressure and velocity of the water in the various parts of the turbine due to the higher head.

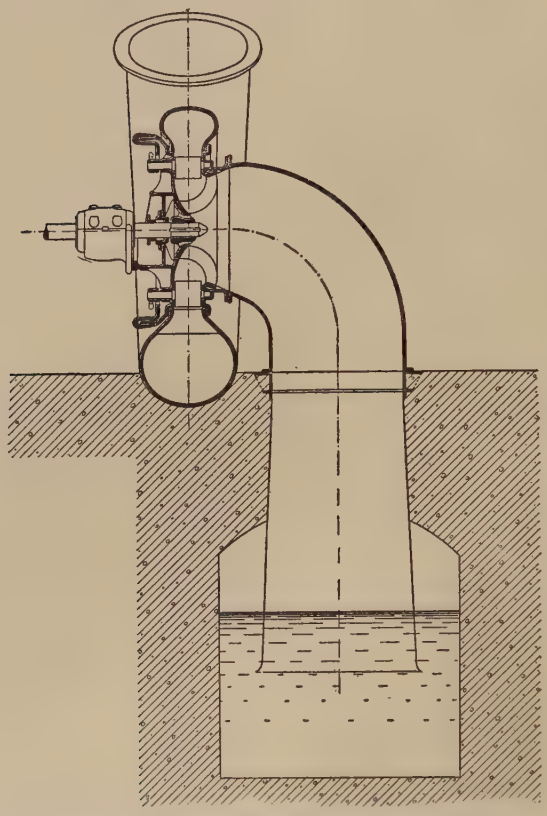

Fig. 178.—Francis Type Turbine with Cast-iron Spiral Casing for High Heads.

As an example of an installation of Francis turbines of this type, we may consider briefly the equipment of the hydro-electric power station at Boquilla-Talsperre, on the Rio Conchos, in Mexico, in which there are six large double units each developing 10,000 h.p. under a head of 61·5 metres (202 ft.) and having a speed of rotation of 360 revs. per min. The general arrangement of the turbines, which were designed and constructed by Messrs. Escher Wyss, will be understood from a study of Fig. 179, which shows a longitudinal section taken along the turbine shaft. It will be noticed that the unit comprises two absolutely similar turbines arranged the opposite way round so that they both discharge into a common central chamber. Each wheel works in a cast-iron spiral casing of approximately circular section, the water being fed into the casing through a large flanged connection which is arranged beneath the turbine as shown in Fig. 180, which is a reproduction of a photograph of the turbine taken after it had been erected in the shops of the makers. The arrangement for controlling the position of the guide vanes is clearly shown, and it will further be noticed that the main control rods are operated by the centrifugal governor arranged at the centre of the unit.

Somewhat different in arrangement, but even larger in size, are the four units, also by Messrs. Escher Wyss, installed at the Ebro Irrigation and Power Co.'s hydro-electric power station at Seros, in Spain. These units, which are of 14,500 h.p. each obtained from a single turbine wheel, are arranged with vertical shafts as shown in Fig. 181, which is a sectional drawing of the turbine, the generator being shown in outside elevation above. The turbines work under a head of 51 metres (167 ft.), the water being fed to the spiral casing through a pipe from the forebay which, together with the necessary screens, has been erected at the top of the hill overlooking the power station. These pipes are no less than 3 metres (9 ft. 10 in.) diameter and are arranged on the hillside at an angle of about

45 degrees, and they are built up of steel plates of thickness varying from 10 mm. (0·4 in.) at the top to 12 mm. (0·47 in.) at the bottom. The spiral casing is also built up of steel plates riveted together, as will be seen from Fig. 182, which is a view of the complete turbine as erected in the works of the makers. The main step bearing, as will be noticed from this view, is arranged immediately above the turbine, its form being shown in section in the drawing (Fig. 181). This bearing is lubricated under a pressure of 25 kg. per sq. cm. (354 lb. per sq. in.). The horizontal shaft, also seen in the drawing, is the

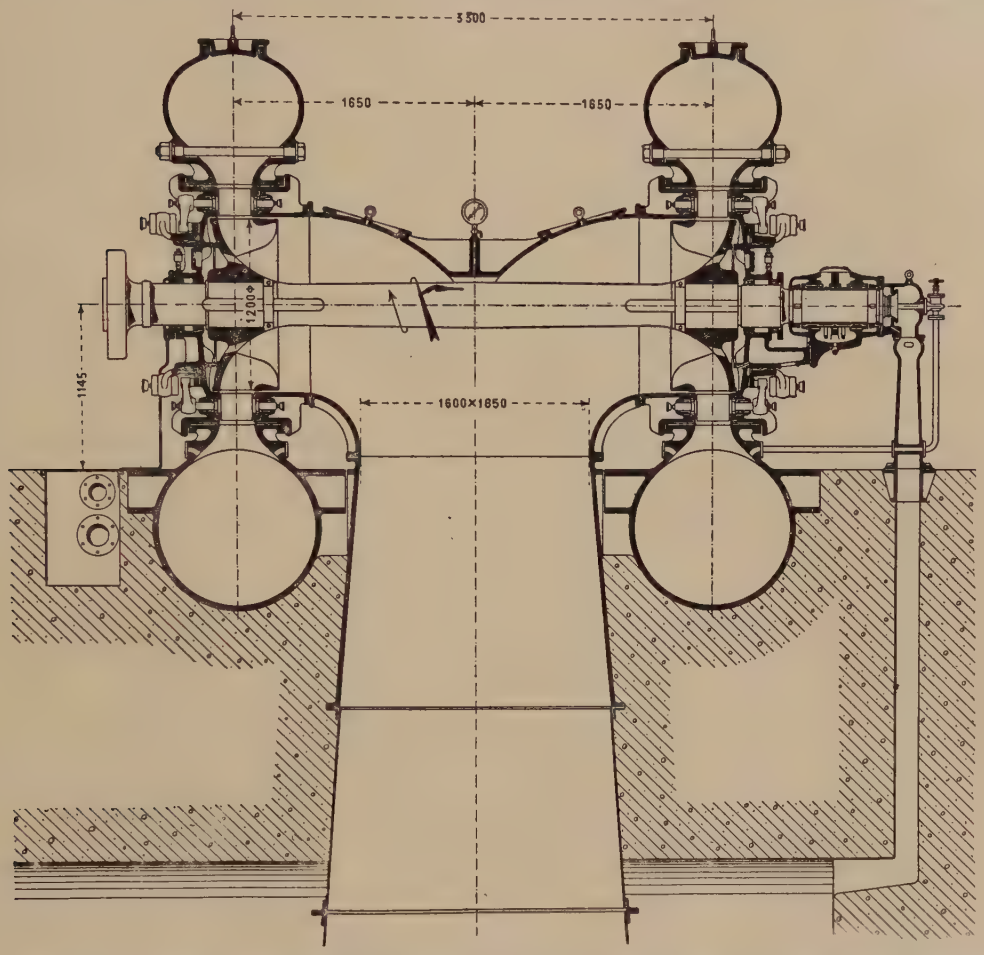

FIG. 179.—Double Runner Francis Turbine of 10,000 h.p., with Cast-iron Spiral Casings, working under a head of 202 ft. Designed and constructed by Messrs. Escher Wyss and installed on the Rio Conchos, Mexico.

control to the guide vanes. The turbine wheel has a diameter at its largest part of 2·13 metres (7 ft.) and has a normal running speed of 250 revs. per min. The alternators, supplied by the General Electric Co., are of 8,000 kw. capacity, three-phase current being generated at a pressure of 6,000 volts and at a frequency of 50 cycles per sec. The current, immediately on leaving the alternators, is transformed up to a pressure of 110,000 volts, at which it is transmitted through overhead lines to the various localities where it is to be used and where, of course, it is again transformed, this time down to a suitable working voltage. Direct current for exciting the four large alternators is supplied from two small direct current dynamos driven by vertical water turbines of 600 h.p. each.

Large as the above unit must be considered particularly in comparison with the different installations to be found in this country, it is far from being the largest in the world. To find these it is appropriate that we should have to go to the Niagara Falls, which is always regarded as the embodiment of Nature's gift of water power. At the time of writing, the largest Francis type turbines in the world are to be found at the Queenston Power Station of the Hydro-electric Power Commission of Ontario, on the northern side of the Niagara River below the Niagara Falls. There will ultimately be in this power station nine units each of 55,000 h.p. working under a head of 305 ft., but so far only five of these have been laid down. In order to obtain the necessary water for operating these turbines, it was necessary to deepen and widen the Welland River between Chippewa and Montrose, a distance of four and a half miles, to construct a canal

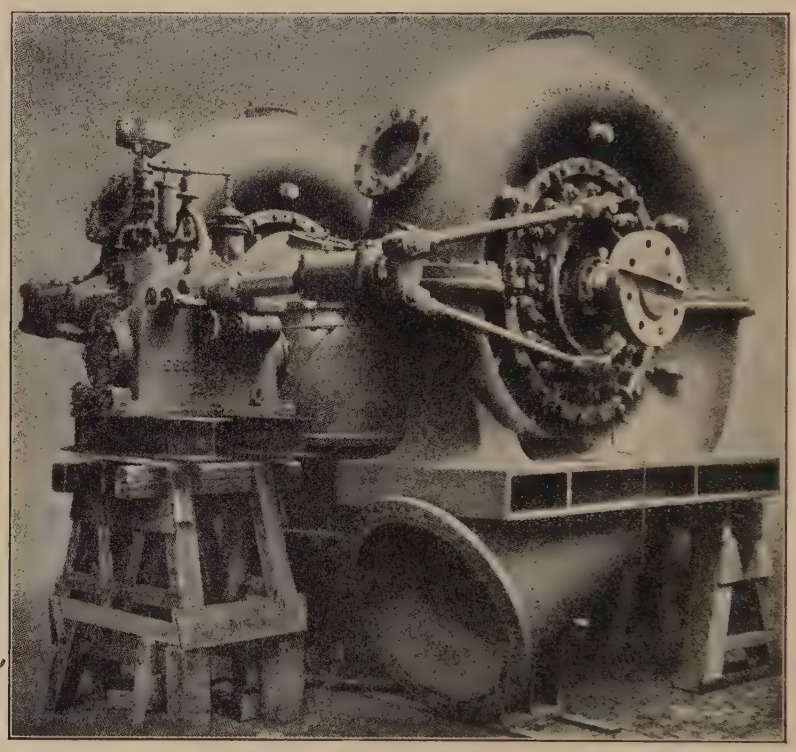

FIG. 180.—Double Runner Spiral Casing Type Francis Turbine of 10,000 h.p. erected in the shops of the makers, Messrs. Escher Wyss et Cie.

eight and a half miles long from Montrose to the forebay and screenhouse at a point on the cliff about a mile south of the village of Queenston, the power house being in the gorge immediately below the forebay. Thus in order to obtain the desired fall, the water is led along a path about thirteen miles long from a point well above the Niagara Falls to a point well below it. A bird's-eye view of the district, showing the path of the water and the position of the power station relatively to the Niagara Falls, is reproduced in Fig. 183, from which some idea of the nature of the work involved can be obtained. The position of the power house is shown in Fig. 184, from which it will be seen that from the forebay the water passes through large built-up pipes to the turbines below. The turbines themselves are of the vertical shaft type, and their action is precisely the same as that of the smaller units already described. The water passes from the forebay through pipes to the turbines below, the spiral casings (or scroll casings, to use the American

term) of which are made of cast steel. The water then passes through the guide vanes to the wheels, the position of the guide vanes being automatically adjusted in order to maintain a constant speed of the wheel despite the variations in the load on the generator. A sectional view of one of these units is shown in Fig. 185, the pipe supplying

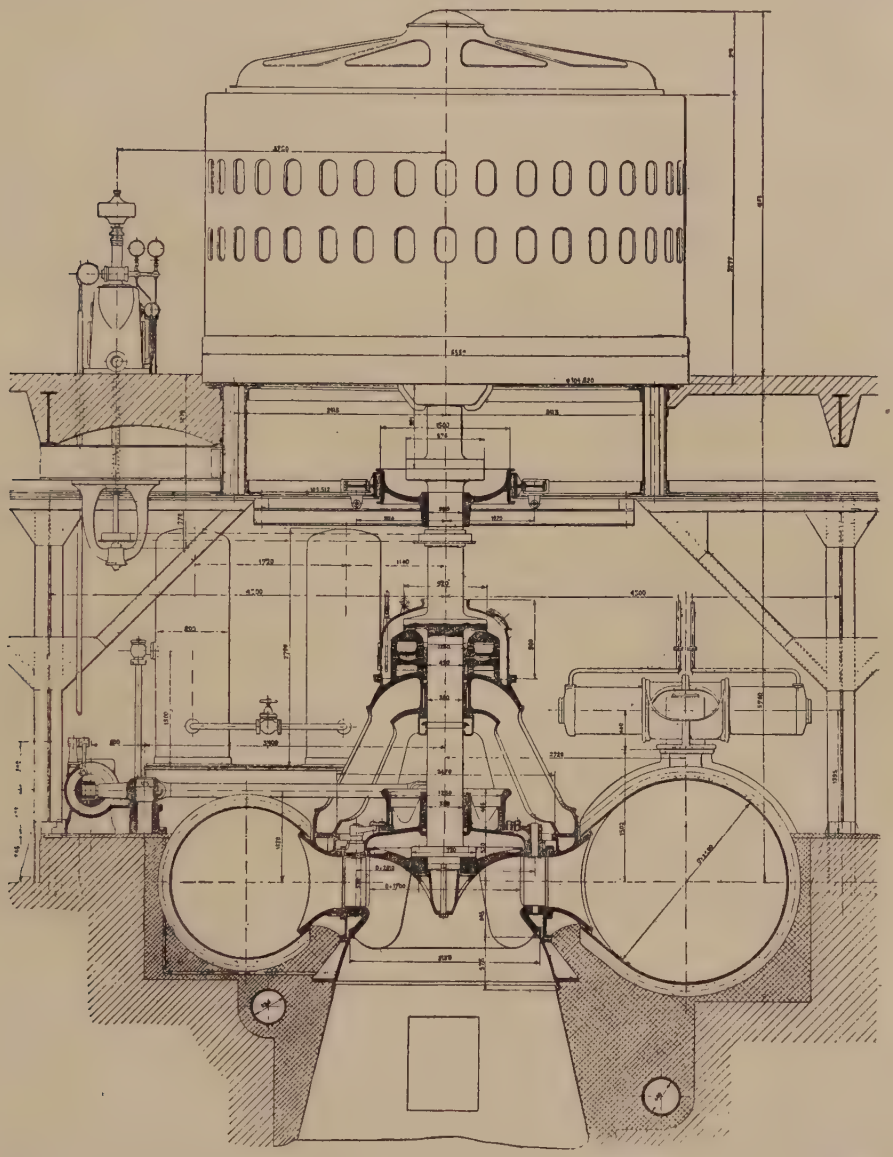

FIG. 181.—Sectional Elevation of Francis Turbine of 14,500 h.p. working under a head of 167 ft. Designed and constructed by Messrs. Escher Wyss for the Ebro Irrigation and Power Co., Spain.

water to the spiral casing being on the left. The man in the galley gives some idea of the dimensions of the various parts of these units and their foundations.

The alternators are also, it is claimed, the largest and the heaviest in the world. These weigh 690 tons each, this being due not only to the large capacity, but also to the fact that the speed of rotation is comparatively slow, being only 187½ revs. per min.

The weight of the revolving parts is carried on a thrust bearing mounted in a housing above the main generator and below the exciting generator, which is at the top, and it is of interest to note that this bearing has been designed to support a load of 450 tons, which is, however, far in excess of the load it is normally required to carry. Current is generated at a pressure of 12,000 volts and at a frequency of 25 cycles per sec., and immediately on leaving the generators it is transformed up to 110,000 volts, at which pressure it is transmitted to the various localities in the province of Ontario where it is to be used. It is interesting to note that despite the enormous capital cost of the station due to the work, not only at the power station itself, but also on the canal and the river, estimated to have been increased by $30,000,000 as a result of the increase of prices following the war,

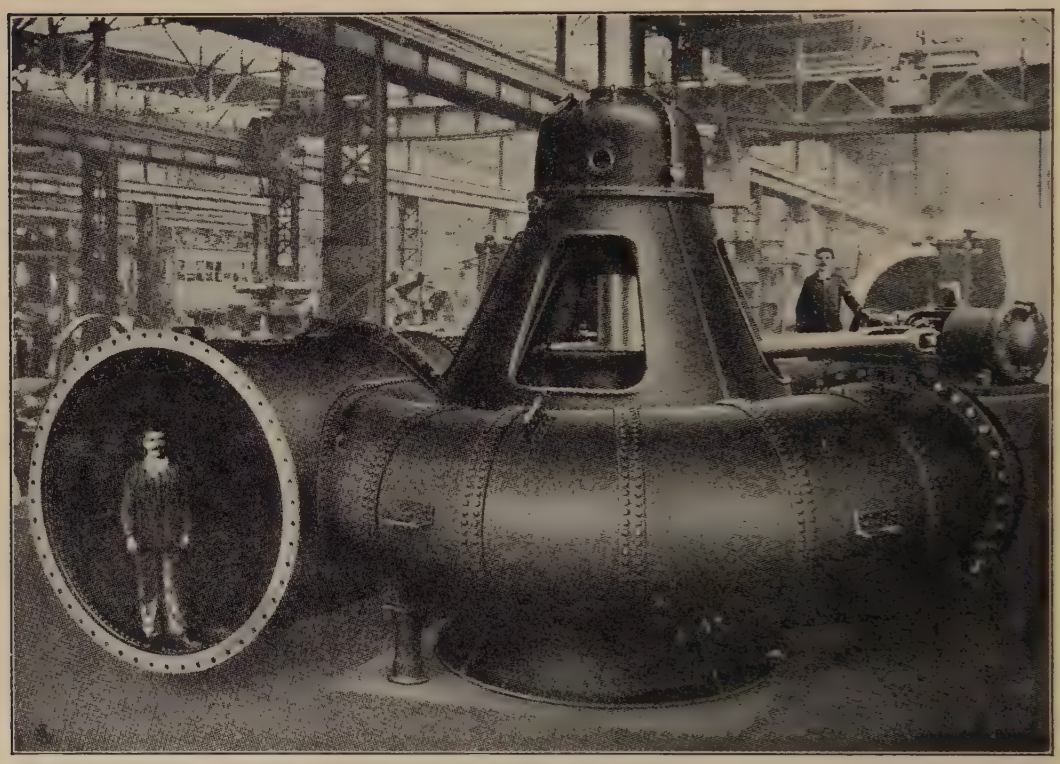

Fig. 182.—Francis Turbine of 14,500 h.p. erected in the works of the makers, Messrs. Escher Wyss et Cie. The Spiral Casing is built up of Steel Plates.

the cost of the electric power supplied on a commercial basis compares very favourably with that in any similar industrial area in the world.

It has already been noted that, for low and medium heads, the Francis type of turbine in one or other of the different forms described above is nearly always employed. In view, however, of the very great commercial importance which hydro-electric engineering has attained during recent years, it is only to be expected that many attempts would be made to devise improved forms of Francis turbines, or even entirely new forms of low-head turbines, with a view to overcoming, if possible, certain of the disadvantages which are inherent to this type. The reader will probably have noticed that where low heads are being dealt with the speed of rotation of the wheel is so small that step-up gearing has to be employed in order to enable the generator to be driven at an economical speed. It is obviously of great value to be able to employ low heads at high efficiency,

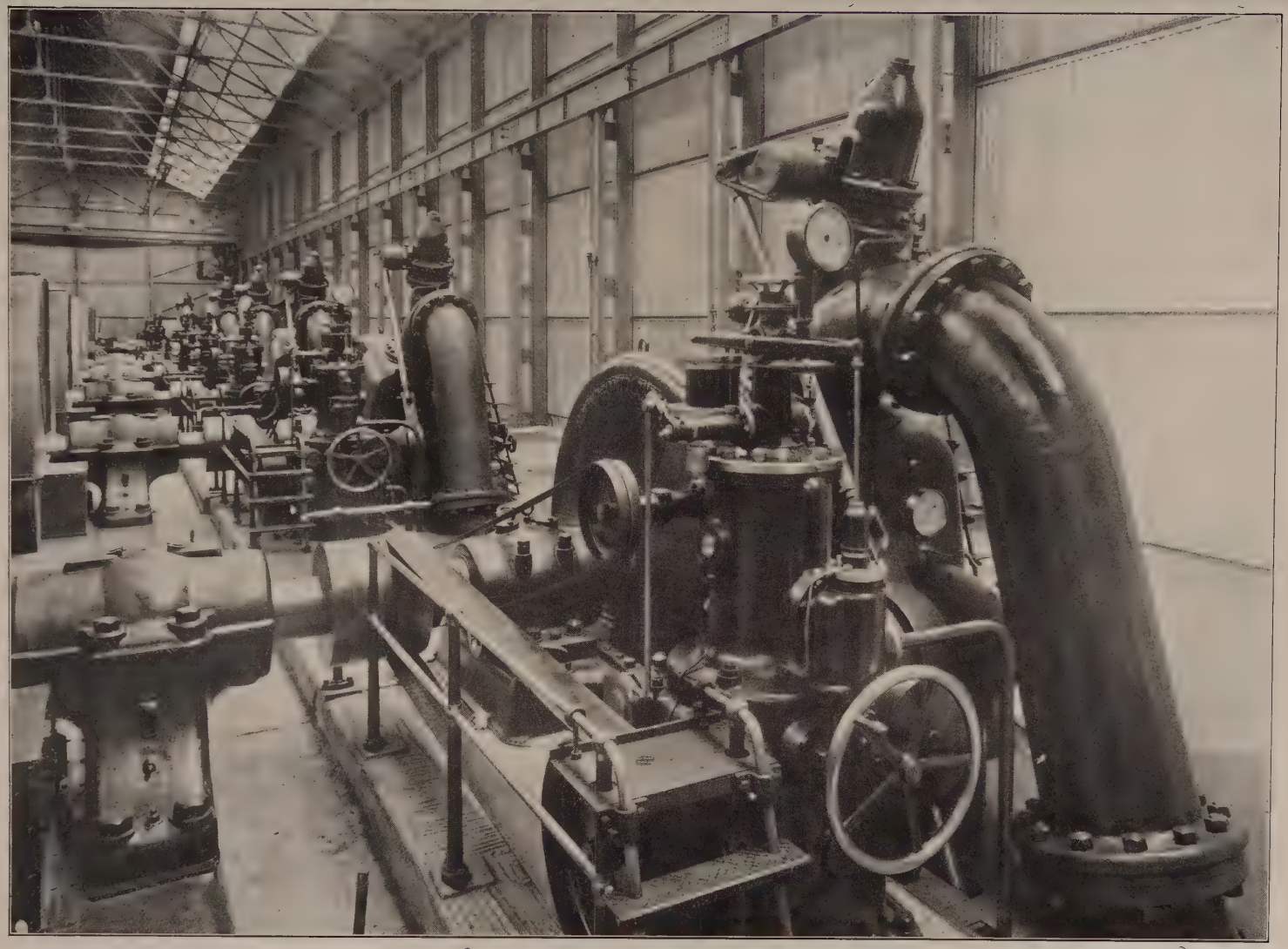

PLATE VIII.—THE POWER STATION OF THE GREAT LAKE HYDRO-ELECTRIC POWER SCHEME, TASMANIA. THE EQUIPMENT COMPRISES FIVE PELTON TURBINES OF 8,000 H.P. EACH AND TWO OF 5,000 H.P. EACH, ALL BY MESSRS. BOVING & CO., LTD., DIRECT-COUPLED TO ELECTRIC GENERATORS. THE LARGER UNITS ARE IN THE FOREGROUND.

and hence a number of attempts have been made to devise a turbine which shall have a higher speed of rotation for a given head while at the same time obtaining a high efficiency, and of these the "Kaplan" turbine is probably the best known. As a result of many experiments in which, among other things, glass tubes were employed in

FIG. 183.—Bird's-eye View of the Niagara River and Falls, together with the Welland River and Canal which supplies water to the Queenston Power House of the Hydro-electric Power Commission of Ontario.

order that the nature of the flow of water from the turbine might be visualised, Professor Kaplan, a Swedish engineer, came to the conclusion that the area of the blades could be very considerably reduced, so that, instead of employing a wheel of the usual Francis type, he gradually evolved a wheel to which there were only four "blades" secured to a small central boss or hub, but without any external ring at all. We thus get a wheel which is very similar in shape and form to a

ship's propeller, and the term " propeller type " is now often applied to this form of turbine wheel. Further, in view of the fact that, with low heads, a small variation in head makes an appreciable difference to the velocity of the water passing through the wheel, Professor Kaplan decided to make the blades of the wheel capable of being rotated so that their slope relatively to the plane of rotation could be changed. If the reader will glance back to Fig. 166 he will at once realise that, if the velocity V_1 with which the water enters the wheel is changed, the speed of the wheel V_2 remaining constant, then V_3 will be changed and with it v_1, the absolute velocity with which the water leaves the wheel. If the direc-

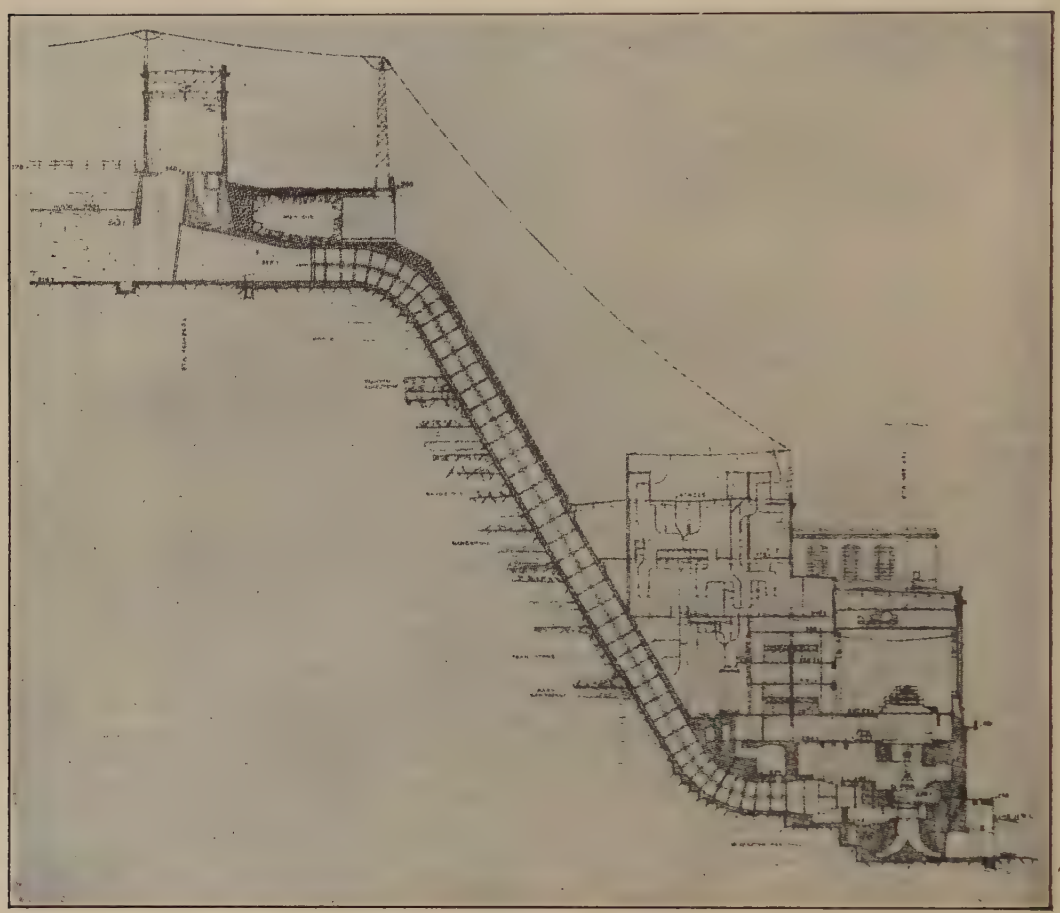

FIG. 184.—The Queenston Power House, showing Forebay above and Pipe Line to Turbines. The High-tension Transmission Lines are also shown.

tion of v_1 changes such that the water, instead of moving axially along the draught tube, is discharged from the wheel inclined to this axis, whirling and eddying will occur with corresponding loss of efficiency. Hence in the Kaplan turbine, not only the position of the guide vanes around the wheel, but the position of the blades on the wheel itself is controlled by the governor in such a way that, no matter what the head may be, the absolute velocity with which the water leaves the wheel is always in the axial direction. This, it is found, leads to a considerably higher efficiency at varying heads than is possible with the ordinary Francis type. A photograph of the " wheel " or runner of a Kaplan turbine is reproduced in Fig. 186.

The general form of the Kaplan turbine will be understood from a study of Fig. 187, which is a sectional elevation of a turbine of 10,000 h.p. designed and constructed in Sweden. We are indebted for this drawing to Messrs. Boving & Co., Ltd., the well-known water power and hydraulic engineers, who are the agents for the Kaplan turbine in this country. It will be noticed from the drawing that the wheel is provided with only four blades which are carried in circular flanges capable of being rotated by a mechanism

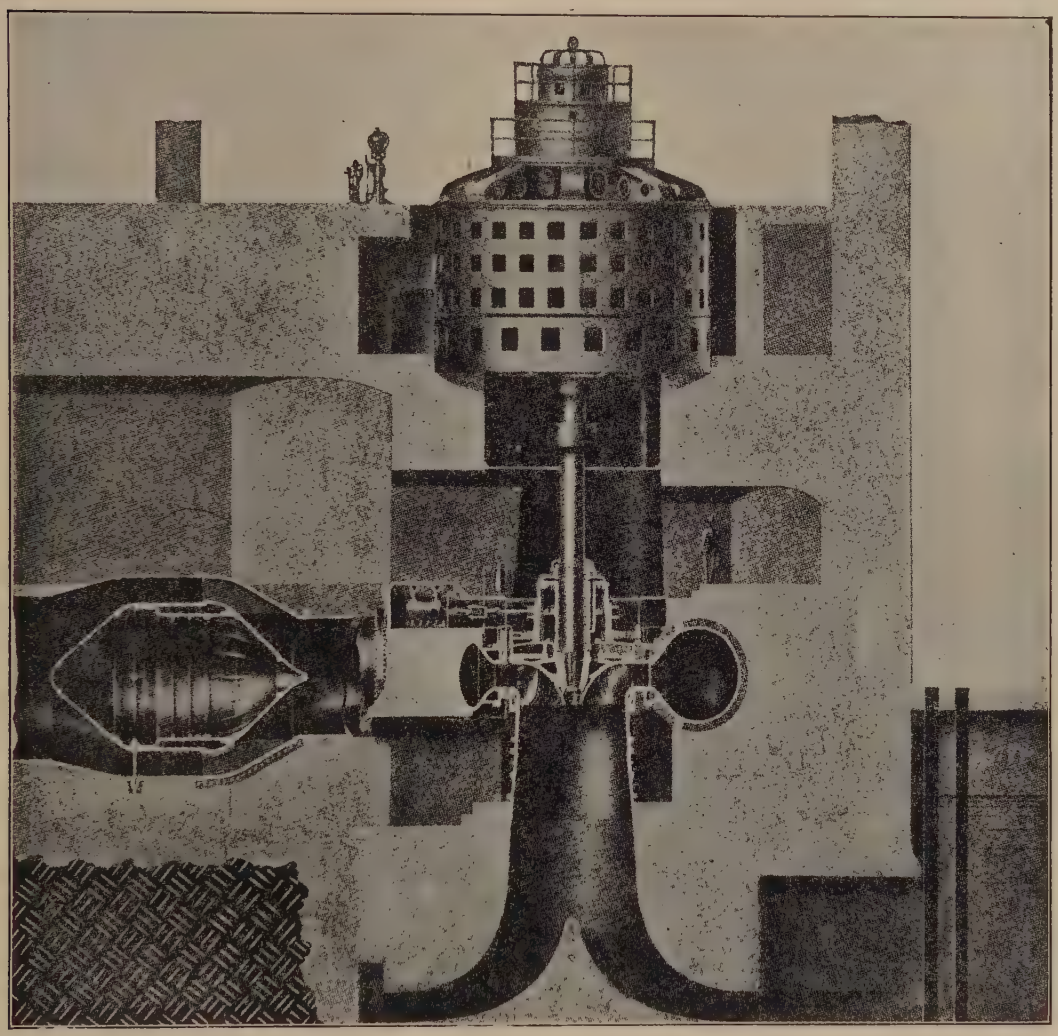

FIG. 185.—Francis Turbine of 55,000 h.p., working under a Head of 305 ft., installed at the Queenston Power Station of the Hydro-electric Power Commission of Ontario.

arranged inside the hollow shaft and controlled from the top by the governor gear as shown. The turbine works under an average head of only 6·5 metres (21 ft. 4 in.) and the speed of rotation in 62·5 revs. per min.; at which speed the generator is directly driven by the turbine. It is of interest to note that a higher speed of rotation could have been obtained if required, but in this case the Swedish Royal Waterfall Board, for whom the turbine was designed, did not desire a higher speed than 62·5 revs. per min. The turbine wheel is 5·8 metres (19 ft.) diameter at its largest part, a dimension which compares very favour-

ably with a Francis wheel of the ordinary type for the same power. The other features of the turbine, including the adjustable guide vanes, do not call for further comment, as they follow the usual practice for Francis type turbines. There appear to be distinct possibilities in the Kaplan turbine, although as far as the author is aware no installation has yet been laid down in the British Isles.

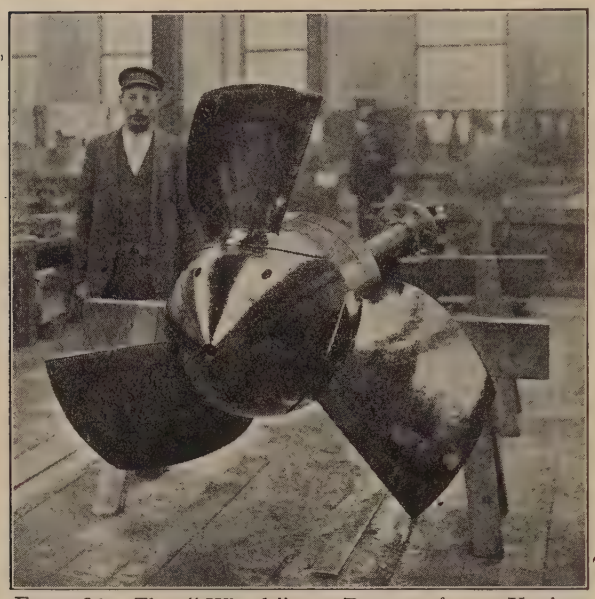

FIG. 186.—The "Wheel" or Runner for a Kaplan Propeller Type Turbine. The Inclination of the Blades is adjusted by the Governor.

We come now to the Pelton waterwheel, the type which is invariably employed where water at a high head is available. The general features of the Pelton wheel have already been explained, but before describing one or two actual installations in detail, it will be of interest to study briefly the theory underlying the action of this important type of water turbine. It can be shown that, for maximum efficiency, the peripheral speed of the blades or buckets of an impulse turbine should be one-half of the velocity of the jet of fluid which impinges upon them. This gives us the normal running speed of the Pelton wheel, the velocity of the jet depending, of course, upon the head, while the actual speed of rotation of the wheel will depend upon its diameter. Thus if V = velocity of jet in ft. per sec., $\frac{V}{2}$ = peripheral speed of wheel. Hence if D = mean diameter of wheel measured to the point at which the jet of water strikes the buckets, and N revs. per min. is the speed of rotation of the wheel, we have

$$\frac{V}{2} = \frac{\pi DN}{60}, \quad \text{or} \quad N = \frac{60V}{2\pi D} = \frac{30V}{\pi D}.$$

The behaviour of the water as it strikes the buckets will be readily understood from a study of Fig. 188, which shows a section of a bucket of a Pelton wheel moving at a velocity $\frac{V}{2}$, the jet of water having a velocity V. It is apparent that the jet of water strikes the bucket at the point A with a velocity relatively to the bucket of $V - \frac{V}{2} = \frac{V}{2}$. The water is then deflected round the bucket, half to one side and half to the other, and neglecting friction, it leaves the buckets at points B, also with a velocity $\frac{V}{2}$ relatively to the bucket, the inclination of the extreme edge of the buckets being usually about 20 degrees to the plane of rotation of the wheel. But the point B is also moving with a velocity $\frac{V}{2}$, so that the absolute velocity of the water leaving the bucket will be V_R, the diagonal of the parallelogram drawn as shown. Thus the water leaves the wheel with a very small velocity in a direction which carries it clear of the wheel, whence it can fall into the pit below the wheel without interference with the water in the next bucket. Theoretically it would

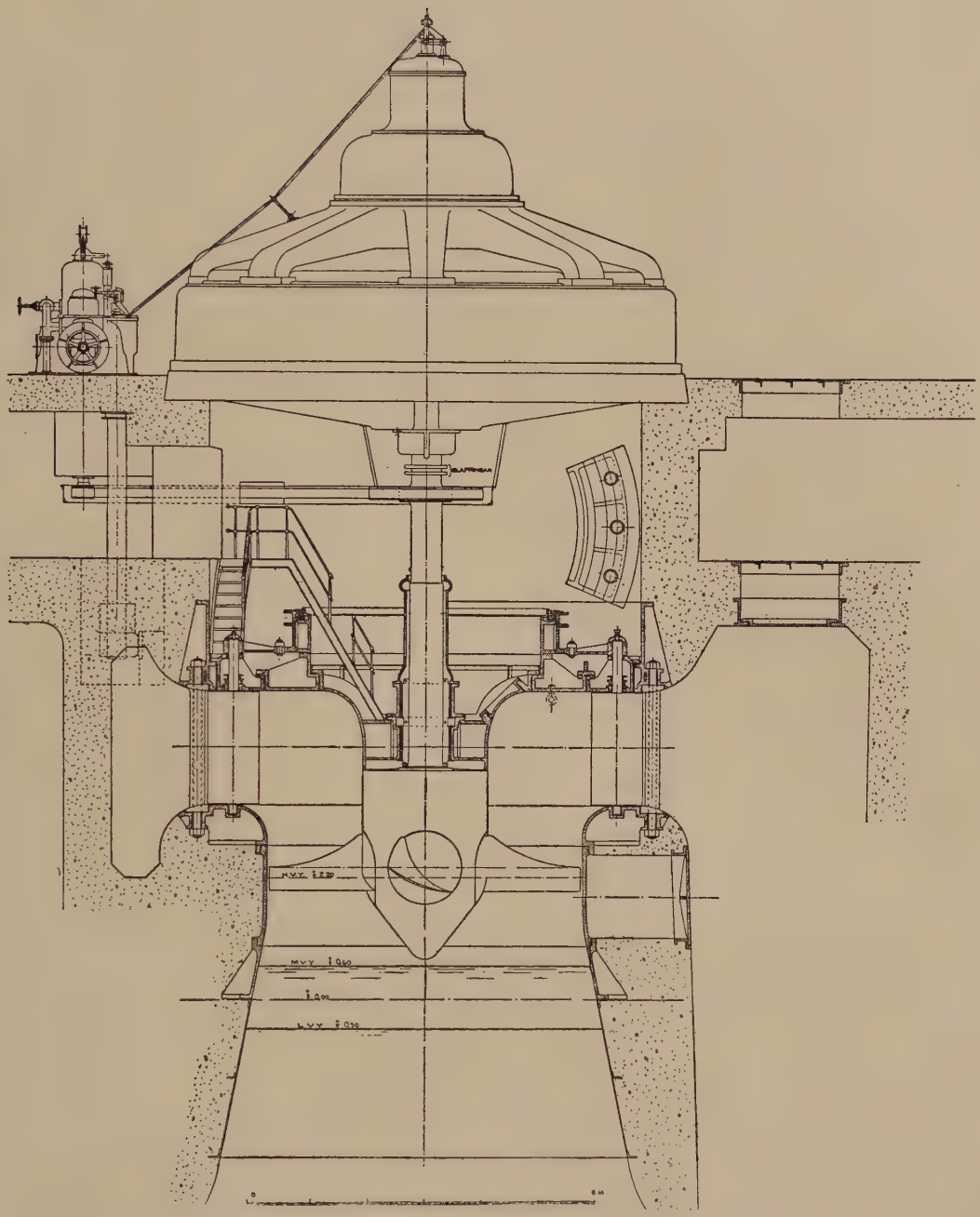

FIG. 187.—Kaplan Propeller Type Turbine of 10,000 h.p. working under a Head of 21·4 ft. Speed of Rotation, 62·5 revs. per min. The Wheel is 19 ft. diameter.

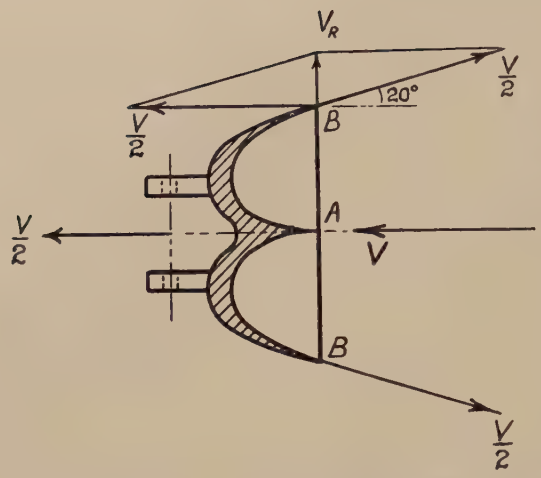

FIG. 188.—Diagram showing Change of Velocity of Water Striking the Buckets of a Pelton Waterwheel.

be advisable to make the angle at B zero, then the resultant velocity V_R would be zero; but then the water would not clear the wheel as soon as it has given up its energy in the bucket, and loss of efficiency would occur due to interference. If, however, we neglect V_R, we have the whole of the energy of the jet absorbed by the wheel.

Owing to the high velocity at which the water is moving in the pipes leading to the nozzles of the Pelton wheel, particular care has to be taken when varying the supply of water delivered through the nozzle to the jet. The obvious method of governing the Pelton wheel in order to obtain a constant speed of the generator despite variations in the load would be to reduce the velocity of the water flowing through the nozzle by means of a throttle valve. This method, however, is not used because any sudden change of velocity in the pipe would set up intense hammering action which might readily cause fracture. This hammering effect can be easily demonstrated at home if a water tap is available in which the screw-down portion moves easily. Open the tap and then, by a rapid flick of the fingers, suddenly close it. There will be a heavy thud in the pipe behind the tap as the moving mass of water is suddenly brought to rest. Imagine now the supply pipe of a Pelton wheel, far larger in diameter, with the water moving at far greater speed, and it will be realised that the force of the blow due to sudden changes of velocity may be very considerable. To overcome this difficulty, other methods of governing have had to be devised. One of the simplest of these is to place a baffle plate in the stream of water issuing from the jet, so that the water is deflected away from the buckets and falls into the pit without doing

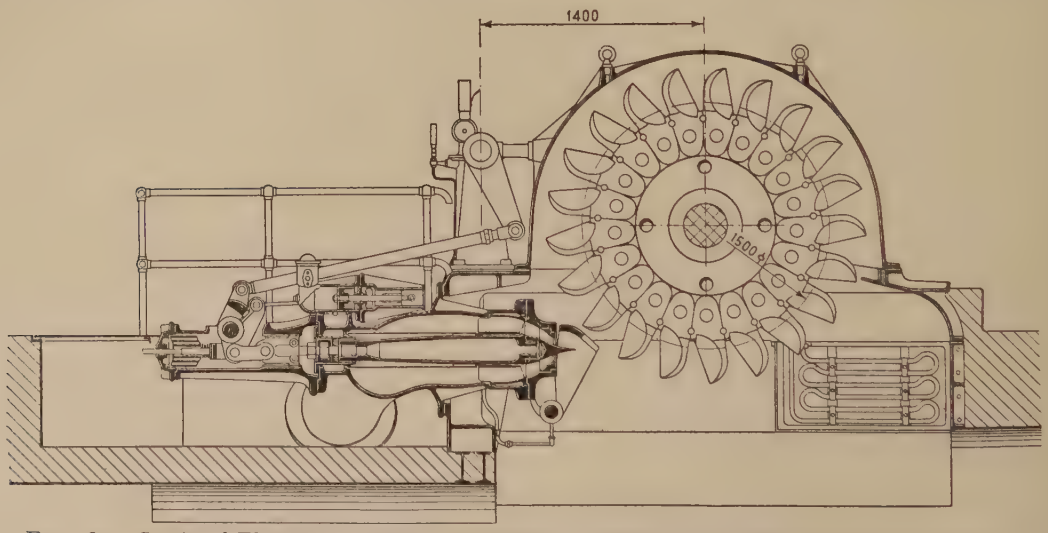

FIG. 189.—Sectional Elevation of Pelton Waterwheel by Messrs. Escher Wyss et Cie, developing 4,200 h.p. at a Head of 1,310 ft.

any useful work. This method is adopted on the turbine shown in Fig. 162 (page 187), the position of the baffle being controlled by the governor, and from the point of view of the security of the pipe lines it yields a particularly simple means of governing the speed of the turbine when the load is liable to fluctuate rapidly over a wide range. It suffers from two disadvantages, however: first, that it does not enable a very sensitive speed regulation to be made; and, second, it is obviously wasteful of water. To get over these difficulties, governing is often effected by means of a central spindle of special form which is brought nearer to or farther from the mouth of the nozzle, thus varying the area of the opening and with it the amount of water in the jet. This method is also incorporated in the drawing referred to above, and in many turbines it will be found that both methods are employed, the former dealing with sudden variations

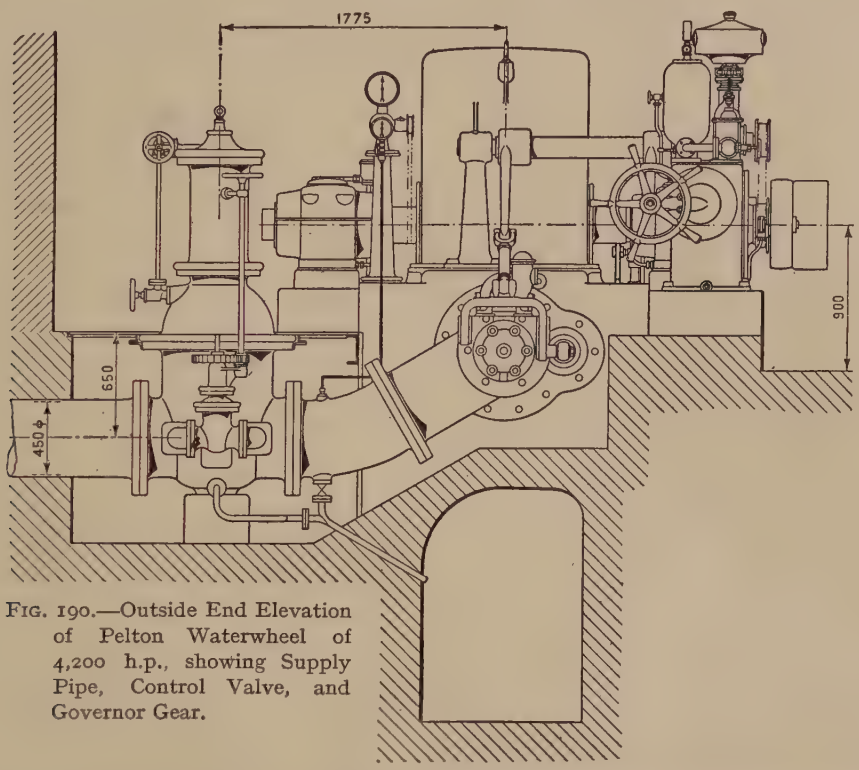

Fig. 190.—Outside End Elevation of Pelton Waterwheel of 4,200 h.p., showing Supply Pipe, Control Valve, and Governor Gear.

of load and enabling the unit to be immediately cut out if necessary without fear of damage to the pipes, while the latter gives very close speed adjustments.

A turbine in which the combined system of governing is employed is shown in Fig. 189. This is a sectional elevation drawing of a Pelton wheel turbine designed and constructed by Messrs. Escher Wyss et Cie for the Hydraulic Power Co., of Moncenisio, Turin, the turbine, which has a wheel 1,500 mm. (59 in.) mean diameter and develops 4,200 h.p., working under a head of 400 metres (1,310 ft.) The spindle in the nozzle and the mechanism for operating it are clearly shown, although the governor, which operates the lay shaft at the side of the turbine casing, is not shown in this view. The baffle plate mounted just outside the nozzle is also clearly shown, although here the operating mechanism is not visible. Attention may, however, be drawn to Fig. 190, which is an end elevation drawing of the same turbine. The centrifugal governor is seen to the right of the turbine casing, driven by a belt from the main turbine shaft. The governor controls the position

FIG. 191.—The Dam, 84 ft. High at the Centre, and Reservoir of British Aluminium Co.'s Hydro-electric Installation.

of the lay shaft, the actual movement of the lay shaft being performed by a servo-motor the operation of which is controlled by the governor. The links operating the needle, as well as the connection to the baffle plate, are clearly shown. The main stop valve which controls the supply of water to the nozzle is also shown in this drawing. Reverting again to the sectional elevation drawing, the lubricating oil cooler placed in the turbine casing may be noted, as well as the oil pipe to the baffle bearing. All the bearings are lubricated under pressure, the oil being led to the different bearings, after which it drains to a sump, is filtered and cooled, and then used over and over again. The interior of the wheel casing is, of course, only subjected to atmospheric pressure, which accounts for the light construction which is required for the casing of the Pelton wheel, this being in great contrast to the spiral casing of the high-head Francis turbines. To guard the pipe line against the effect of shock during the operation of the speed controls, and also when shutting down the unit by means of the sluice valve, various protection devices are usually fitted.

The best-known installation of impulse type turbines in the British Isles is to be found at Kinlochleven, in Scotland, where the British Aluminium Co., Ltd., have a hydro-electric power station in which current is generated for use in the production of aluminium from its ores by the electrolytic method. There are many localities in the Scottish Highlands where considerable amounts of water power at high head are available, but the difficulty is that they are generally found in the bleak and rugged districts far removed from the towns and other large industrial areas. Many schemes have been brought forward for harnessing this power, utilising it to drive electric generators, and then transmitting the energy at high voltage to the districts where it can be used. The cost of transformation at each end of the line, the cost of the transmission lines, and the losses of energy involved in transformation and transmission are very serious items which often affect considerably the commercial feasibility of the scheme. The British Aluminium Co. obviated these difficulties by erecting their works at the site of the hydro-electric power station,

FIG. 192.—Penstock Chamber with Measuring Weir at base of Dam, British Aluminium Co.'s Hydro-electric Installation.

the aluminium works being immediately adjacent to the power house, so that transmission losses are practically zero. Kinlochleven is situated at the head of Loch Leven, a branch of the larger Loch Linnhe, and beneath the shadow of Ben Nevis, a mountain 4,406 ft. high. The valley of the Leven rises rapidly from the head of the Loch to a plateau containing a chain of three small lakes and embracing a large area in its watershed, which includes a considerable portion of Rannoch Moor. A dam has been constructed at a point five miles from the end of Loch Leven to combine these lakes into a large storage reservoir. From this reservoir, water is conveyed through a conduit three and a half miles long to an open penstock chamber on the hillside overlooking the factory; it then passes from the penstock through pipe lines to the turbines below, where a head of 925 ft. is obtained. The watershed encloses an area of 55 sq. miles on which the average annual rainfall is about 84 in., and the amount of water thus provided at the available head is sufficient for a continuous full load of at least 22,000 h.p. throughout the year.

FIG. 193.—Conduit, 8 ft. Wide by 8 ft. Deep, conveying Water from the Reservoir to the Pipe Line, a distance of three and a half miles, British Aluminium Co.'s Hydro-electric Installation.

While this is very far from being a large hydro-electric undertaking, some idea of its magnitude will be gathered from the fact that the dam, which is nearly three-quarters of a mile long, is 84 ft. high at the centre and has a maximum width at the base of 63 ft. tapering to 10 ft. wide at the crest. The dam together with a portion of the reservoir is shown in Fig. 191, from which view a good idea of the rugged nature of the country can be obtained. Water is taken from the reservoir at the base of the dam by three 42 in. pipes fitted with double independent sluice valves which are controlled from a valve tower erected on the top of the wall. The outer valves serve to regulate the flow of water, while the inner valves, which are normally kept open, are held in reserve to allow the outer valves to be dismantled and repaired when they become worn by the continuous rush of water. The water then discharges freely into the penstock chamber shown in Fig. 192, where it is led over a knife edge measuring weir before entering the conduit. Sluices are provided as shown, to draw off the whole of the water from the penstock if desired. The conduit, which is of concrete of square section 8 ft. by 8 ft., follows the contour of the hillside, although in one or two places viaducts have had to be erected to carry it over

FIG. 194.—Lower Penstock and Valve House at Head of Pipe Line, British Aluminium Co.'s Hydro-electric Installation.

narrow gorges in the hillside. This conduit has a slope of about 1 in 1,000, and serves to convey the water to the lower penstock and valve house at the head of the pipe line, the water flowing at a maximum depth of 7 ft. 6 in. in the conduit. A view of the conduit showing the nature of the country through which it passes is given in Fig. 193, and it may be noted here that at three points along its length where side streams are crossed their water is turned into the conduit.

The lower penstock which, together with its valve house, is shown in Fig. 194, is a structure 140 ft. long, 52 ft. wide, and 15 ft. deep built in concrete partly reinforced. It is provided with a 1 in. grating with breakwater arches to check the flow of water coming from the conduit, with a knife edge measuring weir for observing the discharge at any moment, and with flushing gates for clearing deposits of rubbish and sand. An overflow serves to maintain the balance between the supply of water arriving from the dam and the quantity passed through the pipe line to the turbines. Owing to the length of the conduit and its small slope, nearly an hour elapses before any variation in the volume of water discharged from the dam affects the supply arriving at the lower penstock chamber, and careful regulation is necessary in order to prevent excessive waste of water and at the same time to obviate the risk of temporary shortage to the turbines. This, however, is not so difficult a matter as it would be for an installation supplying electricity to a town, as in aluminium furnace operation an almost constant current is consumed throughout the twenty-four hours of each day. The pipe line, a view of one portion of which is shown in Fig. 195, consists of six welded steel pipes each of 39 in. internal diameter and of total length 2,100 yards, the difference of level between the penstock and the power house being 925 ft. The pipes are supported on concrete pedestals, heavy concrete anchorages being provided at all bends as shown. The loss of head due to friction is about 3½ per cent., so that the working head at the turbines is 890 ft. A hydraulically operated sluice valve is provided at the bottom of each pipe, there being an automatic closing device to each valve which comes into action if the velocity of the water in the pipe exceeds a specified figure. Throttle valves having similar automatic closing apparatus are fitted at the top of the pipes below the penstock chamber, these being provided in order to protect the works should a burst occur. Tubes open to the air are fitted to the pipe lines below these valves, so that no vacuum may be formed when the pipes are emptied of water. The map reproduced in Fig. 196 gives the exact location of the dam, the conduit, and the pipe line, and serves to show the extensive nature of the work involved in harnessing the water in order to make it available for driving the turbines in the power house at the aluminium factory.

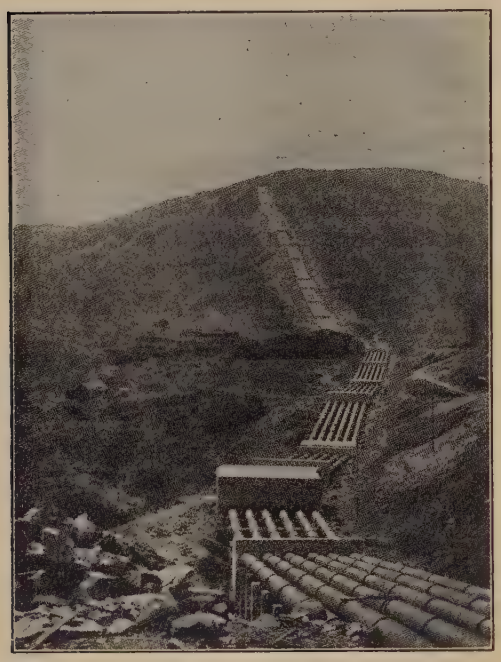

FIG. 195.—Pipe Line, comprising Six 39-in. Pipes, conveying Water to the British Aluminium Co.'s Power Station at Kinlochleven, giving a total Head of 925 ft.

The contract for the water turbines was awarded to Messrs. Escher Wyss et Cie, the construction of the dam, conduit, pipe line, etc., having been carried out by the well-

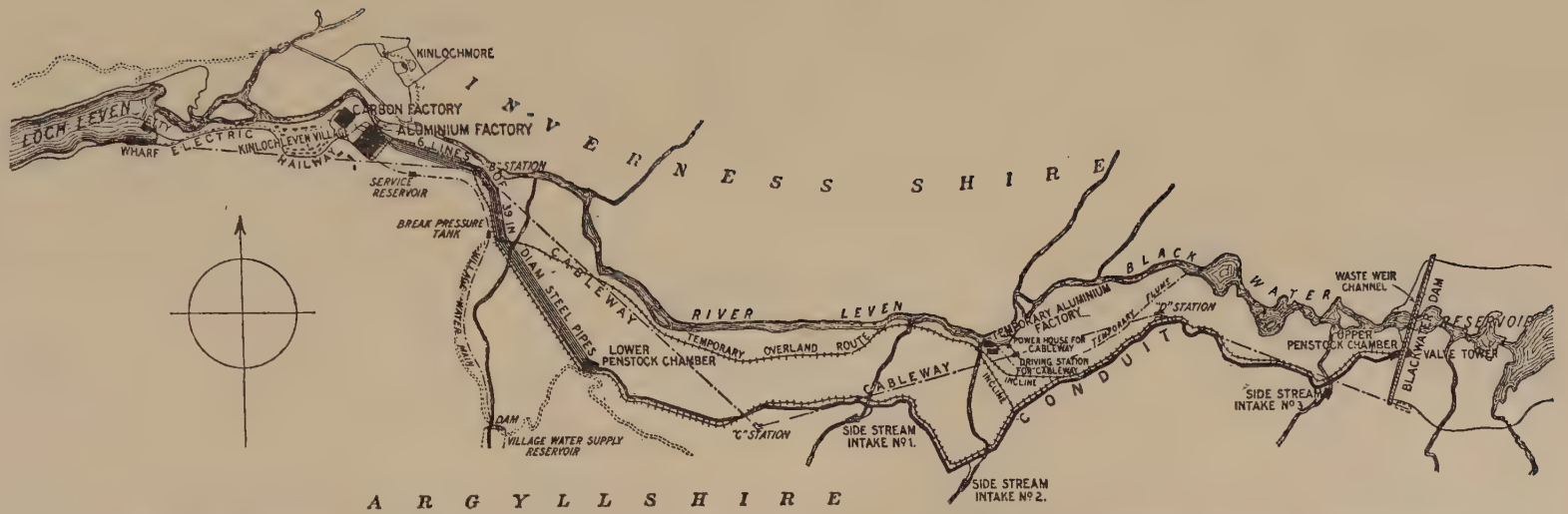

FIG. 196.—Map showing the District in which the Hydro-electric Installation of the British Aluminium Co. has been developed. The Reservoir is on the right, connected by Conduit and Pipe Line to the Power House at Kinlochleven on the left.

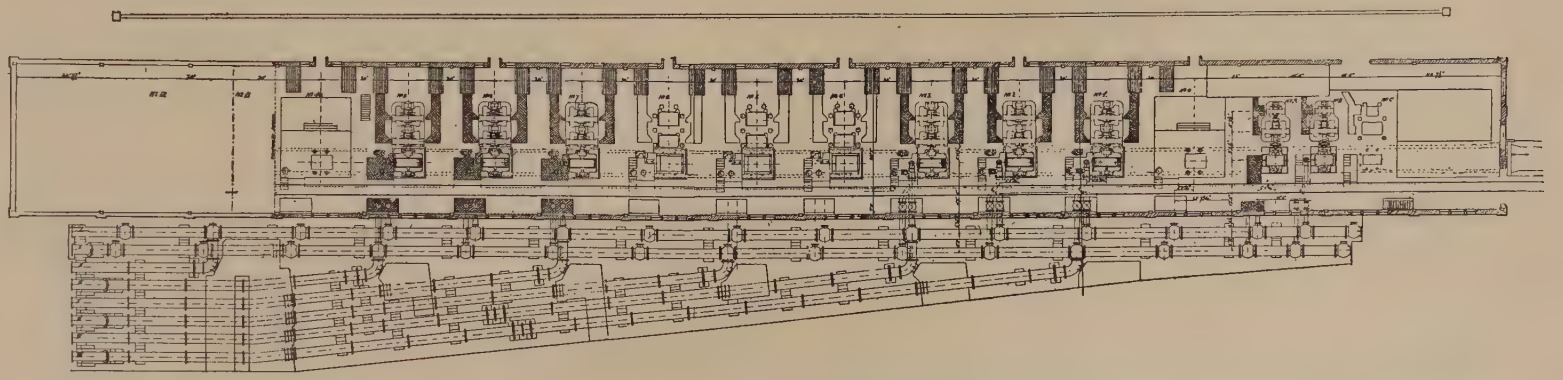

FIG. 197.—Plan showing Water Distribution System and Layout of Hydro-electric Power Station of the British Aluminium Co. at Kinlochleven.

known constructional engineers Sir John Jackson, Ltd. Eleven turbines of the Pelton wheel type were installed, nine of these being of 3,200 h.p. maximum output each, for supplying current to the aluminium furnaces, while two smaller units of 930 h.p. each were provided for supplying light and power to the works and the railway which transports the materials between the works and the jetty at the head of Loch Leven. The general layout of the turbines and the distributing system through which the water is delivered from the pipe line to the turbines is shown in Fig. 197, from which it will be observed that two "bus" pipes, each fed by three of the mains on the pipe line, extend the whole length of the power house, each connecting up to the inlet pipes to all the turbines. The six mains of the pipe line are coupled to the six pipes lying outside the two bus pipes, the connections to the bus pipes being made as shown. The bus pipes are arranged the one 4 ft. higher than the other, so that connection to the turbines can be conveniently made. A view taken from the side of the power house showing the bus pipes and the pipe line in the distance is reproduced in Fig. 198. The turbines are erected over a concrete pit running the entire length of the power house, this constituting the tail race from which the water, after it falls from the buckets, passes into the river.

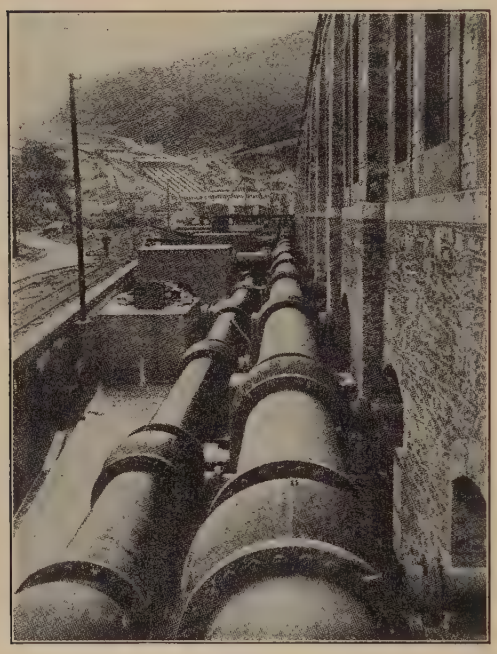

FIG. 198.—View of Side of Power House at Kinlochleven showing "Bus" Pipes with Supply Pipe Line in the Distance.

The general features of the design of the large turbines will be understood from a study of Fig. 199, which shows a side elevation and a longitudinal sectional elevation of the Pelton wheel, together with the nozzles and the control gear. At the effective head of 890 ft., the pressure of the water before it leaves the nozzles is 385 lb. per sq. in., and at maximum load the discharge from the nozzles of the 3,200 h.p. turbines is 42·5 cu. ft. of water per sec. The speed of rotation of the wheel can be varied between 300 and 330 revs. per sec., while the guaranteed efficiency is 79 per cent. at full load, 80 per cent. at three-quarters load, and 77 per cent. at half load. Two circular nozzles each 5 in. diameter direct jets of water on to the buckets of the wheel, the size of the jets being controlled by needles which are operated from the governor, the design of the nozzles and the needles being such that a perfectly compact jet is maintained for all values of the discharge. The nozzles are made of steel castings lined with bronze, while the needles have detachable heads made of hard bronze. The buckets are machined and carefully ground so that a fine cutting edge is presented to the water jet, and their surfaces, as well as that of the needles and nozzles, are highly polished in order to reduce friction losses to a minimum. The buckets are secured to the running wheel by means of steel rivets and taper pins, while the wheel itself is mounted on a steel shaft supported in white metal lined bearings lubricated by means of ring oilers.

The supply of water to each turbine is through valves each 500 mm. (19⅝ in.) diameter fitted to the two inlet branches from the two bus pipes. These are operated by hand from the power house floor and are provided with by-pass valves to relieve the thrust when opening and closing the valves. The turbine inlet pipe formed by the junction of

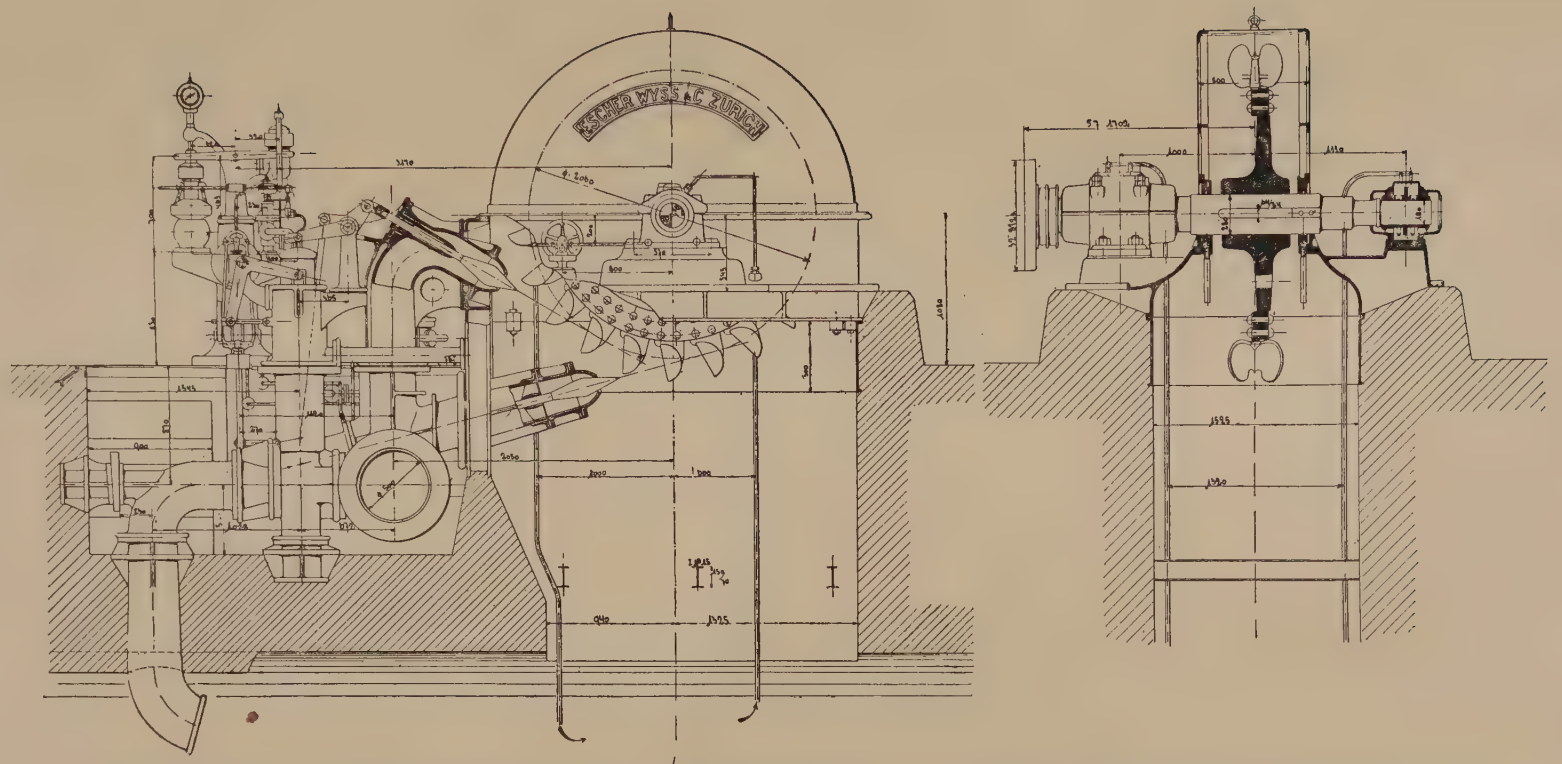

FIG. 199.—Side and End Elevations of Pelton Waterwheel of 3,200 h.p., designed and constructed by Messrs. Escher Wyss et Cie for the British Aluminium Co., Ltd.

these two branches is laid in a trench as shown below the level of the floor, so that the whole of the power house is left unencumbered with pipes. The regulation is effected by Messrs. Escher Wyss et Cie's hydraulic system, in which water under pressure from the mains, i.e. at about 385 lb. per sq. in., is admitted through a relay valve under the control of the governor to a servo-motor mounted on the needle of the lower nozzle. The two needles are, however, rigidly connected by means of levers and connecting rods, so that the upper needle follows exactly the movements of the lower needle. The arrangement of the governor gear and the connecting mechanism between the two nozzles is shown in Fig. 200, which is a view of the turbine as erected, but not coupled up to its electric generators, the floor plates being removed to show the regulating gear. When water under pressure is admitted from the relay to the servo-motor cylinder, the needles open

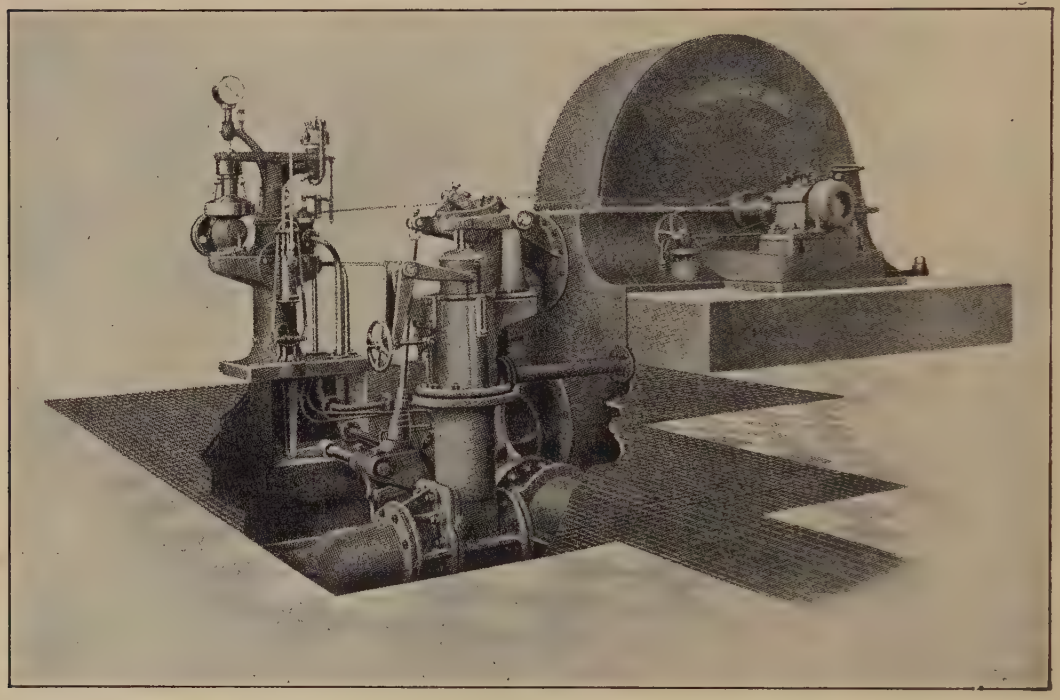

FIG. 200.—Pelton Waterwheel of 3,200 h.p. with Governor, designed and constructed by Messrs. Escher Wyss et Cie for the British Aluminium Co. The Floor Plates are removed to show the Control Gear.

and allow more water to impinge on the buckets. A spring fitted to the end of the needle rod is thereby compressed, and this spring, together with the force of the water on the needles, serves to close the nozzles when the pressure is released from the servo-motor. The governor is of the spring controlled centrifugal type and its action is steadied by an oil brake cylinder. It is driven from the turbine shaft by double belts, so that in the event of one breaking or falling off the pulley, the other is still available to drive the governor. Connection between the governor and the relay valve is by means of levers, so that any slight variation in the speed of the turbine immediately results in the admission or the release of water under pressure to or from the servo-motor, which responds by opening or closing the nozzles a proportionate amount according to the requirements of the turbine. At the same time, the speed of the turbine can be controlled by means of the hand wheel which alters the relative position of the governor and the relay valve spindle. This hand wheel will be seen immediately below the governor. Further, in order to prevent

water hammer shocks from occurring in the pipe line when the turbine is shut down, the Escher Wyss patent pressure relief valve is fitted which opens in proportion as the needles close, thus allowing exactly the same volume of water to escape directly into the tail race, so as not to check the velocity of flow in the pipes. The relief valve then slowly closes, reducing the area through which the water passes, and thus gradually retards the velocity of the water in the pipe. The relief valve and its escape pipe to the tail race are clearly shown in the side elevation drawing of the turbine (Fig. 199).

The nine larger electrical generator sets which are direct coupled to the 3,200 h.p. turbines consist of two identical multipolar shunt wound direct current generators in the same shaft, each of 1,000 kw. capacity at a pressure of 250 to 275 volts mounted on the same bed plate and electrically connected in parallel. It was found convenient to divide these generators into two units on account of the excessive size of the armatures and the commutators which would be required to carry the current, which amounts to 8,000 ampères per set, so that the current in each armature, etc., is by this arrangement reduced to 4,000 ampères. As in the case of the turbines, the generators have been designed to run continuously at full load for long periods. The bearings are provided with forced lubrication, and all brushes are arranged for adjusting, cleaning, and replacing while the machine is on load. The efficient operation of the aluminium furnaces requires a variation of the electrical pressure between 250 and 275 volts, and this is effected by altering the speed of the turbines from 300 to 330 revs. per min.

The smaller lighting and traction sets are driven by turbines having a normal output of 830 h.p., but they are capable of giving 930 h.p. on overload. The turbines, which are of the Pelton type, are of substantially the same design as the larger sets except that only one nozzle is provided to each wheel, the maximum discharge from which is 12·3 cu. ft. per sec. The speed of rotation of the wheels can be varied between 400 and 440 revs. per min., and the guaranteed efficiency is 79 per cent. at full load, 80 per cent. at three-quarters load, and 77 per cent. at half load. These turbines also drive two independent generators on the same shaft, but in this case the generators are not identical. The larger of the two is a direct current shunt wound machine of 500 kw. giving 2,000 ampères at a pressure of 250 volts, which supplies the lighting and the general power circuits in the factory and in the village. It also supplements the power from the main generators in case a series of furnaces runs cold and requires additional current. The smaller generator is a direct current compound wound machine of 75 kw. capacity, which supplies current at a pressure of 500 volts for electric traction on the railway between the factory and the wharf, and also for operating the cranes in the factory and at the wharf.

We have described the installation at Kinlochleven at considerable length because it represents, at the time of writing, the largest hydro-electric scheme which has so far been carried out in the British Isles. It will be realised, of course, that the conditions in a power station supplying electricity to an aluminium factory are somewhat different from those which would obtain in a power station supplying light and power to a large industrial area, particularly as regards the fluctuation of the load during the twenty-four hours of the day and night. But as regards the general principles involved in the storage of the water and its transmission to the turbines, these will be the same in every case, and our description may be taken, therefore, as illustrating the nature of the problems which have to be solved in the working-out of any hydro-electric scheme, although the details must necessarily depend upon the local features in each case. When, however, we come to consider some of the notable hydro-electric installations on the Continent and elsewhere in which high heads are available, we find that turbines of considerably greater power than those at Kinlochleven are employed, and it will be of interest to consider briefly one or two of these.

The Government of Tasmania have been quick to realise the economic advantages

of cheap electric power, and much has been done to develop the abundant water power resources of the island. The largest hydro-electric power station to be found in Tasmania uses the famous Great Lake as the source of supply, this lake, which is situated approximately at the geographical centre of the island, being situated at an altitude of 3,350 ft. above sea level on a central plateau. From the southern end of this lake flows the River Shannon, and parallel to it, but about two miles to the west, is the River Ouse, which falls in the first few miles of its course a great deal more rapidly than the Shannon, with the result that a few miles south of the lake the level of the Ouse is more than 1,000 ft. below the Shannon. It is obvious, then, that a short diversion from the latter river to the former enables a very high head to be obtained, and the scheme has been developed on these lines.

Fig. 201.—Wooden Stave Pipe Line supplying Water to the High Falls Power Station of the Hydro-electric Power Commission of Ontario.

It may be added, as indicating the nature of the work involved in developing a large water power scheme, that while originally the water from the Great Lake alone was used, as the scheme extends a further volume of water will be obtained by diverting the head waters of the Ouse into the lake, and in addition the waters of a river known as the Little Pine will be turned into the Ouse and thence to the lake, thus increasing the total capacity to 70,000 h.p. at the turbine shafts. A large dam has been constructed at the southern end of the lake, raising its level and thus impounding sufficient water to enable the plant to give its full output even during the dryest years which can be expected. Starting from this dam, the water flows down the natural bed of the River Shannon for a distance of five miles, when by means of a diversion weir it is turned into a canal three and a half miles long, leading into a storage reservoir just above the pipe lines of the power station.

This storage reservoir was created by forming a lake in a natural depression of the country by means of three low earth dams. A concrete penstock chamber provided with screens and sluice valves was erected at the side of this artificial lake through which the water is drawn off to the pipe line.

The first two pipe lines which were laid down, it is of interest to note, were made up of wooden staves, this being a cheap and convenient construction which is perfectly safe if comparatively low heads are being employed. Obviously such pipes cannot withstand considerable pressures either due to the head of water in them or to water hammer shocks, but for the first portion of a long pipe line, i.e. where the pressures are low, it is frequently employed. A good idea of the method of constructing a wooden stave pipe line will be obtained from a study of Fig. 201, which shows a portion of the pipe line supplying water to the High Falls Power Station of the Hydro-electric Power Commission of Ontario. Here, as we have seen with other stations in the Niagara district, the fall is comparatively low, and wood stave pipes are sometimes used to deal with the entire fall. The building at the top of the illustration is the sluice valve house erected at the side of the penstock. Returning now to the Great Lake installation, two wooden pipe lines have been provided, one 4 ft. internal diameter and the other 5 ft. diameter, for the first part of the line, these being merely rested on wooden logs and not anchored at all except at their lower ends, where they joint the steel pipes which carry the water down to the power house. These steel pipes are, of course, subjected to higher pressures as the fall increases until when the power station is reached the head is about 1,100 ft. giving a pressure of 476 lb. per sq. in., or, allowing for friction, the net head at the turbines is 1,015 ft. giving a pressure at the nozzles of 440 lb. per sq. in.

The present equipment of the power house consists of seven turbines all of which have been supplied by Messrs. Boving & Co., Ltd., the first two being of 5,000 h.p. each, and the remaining five of 8,000 h.p. each. The smaller units, which have a normal running speed of 375 revs. per min., are direct coupled to alternating current generators each having a capacity of about 3,600 kw., the current being generated at a pressure of 6,600 volts and a frequency of 50 cycles per sec. The larger turbines are provided with two nozzles each, the control of the water delivered to the wheel buckets being by combined needle and deflector plate as already described (see Fig. 189, page 214). The general features of the design of these turbines will be gathered from a study of Figs. 202 and 203, which show respectively the front and the back view of the turbine, while Fig. 204, which is a view of the wheel before erection in the turbine casing, is of particular interest as showing the form of the buckets and the sturdy nature of the construction of the Pelton wheel. The wheel, it may be noted, has a mean diameter of 1,780 mm. (5 ft. 10 in.), while the width of the buckets is 540 mm. (21¼ in.). The larger turbines are direct coupled to electric generators of about 5,800 kw. capacity, these being of the totally enclosed type, as will be observed from a study of Plate VIII, which shows the complete station, a portion of the generators being seen to the left of the water turbines. It may be noted that the two smaller sets, which were the first to be installed, are in the background of the picture, and that space is available for extending the power station in the direction of the observer when required. After leaving the generators, the electric energy is transformed up from 6,600 volts to 88,000 volts, at which pressure it is transmitted through overhead conductors supported on steel lattice towers to Hobart, the chief town of Tasmania, which is sixty-two and a half miles away, and also to other places nearer the power station where the electrical energy is utilised.

In all the different sizes of Pelton wheels which we have so far described, the shaft has been horizontal, this being rather different from Francis turbine practice, where in very many instances the vertical shaft arrangement is adopted. There is much to be said in favour of both the vertical and the horizontal shaft system, but the chief advantage

of the former is that it takes up a far smaller amount of floor space than does the latter for a unit of given power. Hence for units of large output the vertical shaft arrangement is to be preferred, and we find that large Pelton wheels are sometimes constructed in this way. A very good idea of the general arrangement of a vertical shaft Pelton wheel will be gathered from Fig. 205, which shows a turbine of 15,000 h.p. designed and constructed by Messrs. Escher Wyss et Cie for the Biaschina power station at Bodio, in Switzerland. This turbine works under a head of 255 metres (836 ft.), the water being led through two

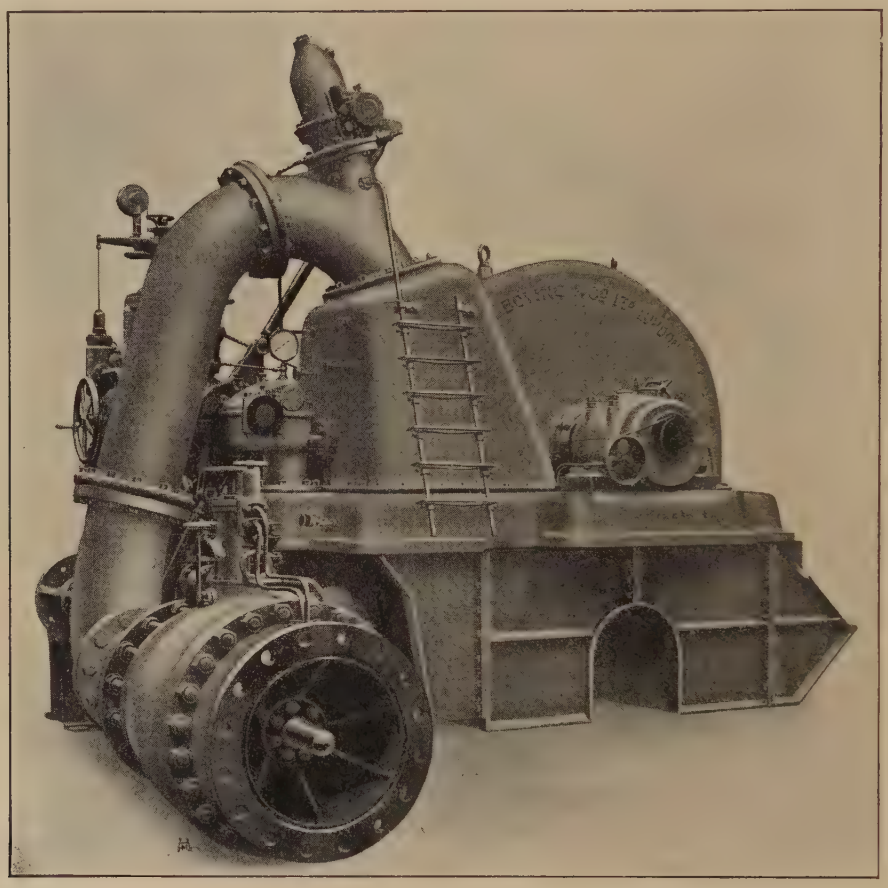

FIG. 202.—Front View of Pelton Wheel of 8,000 h.p., designed and constructed by Messrs. Boving & Co., Ltd., for the Great Lake Hydro-electric Scheme, Tasmania.

pipes to the turbine, each of these pipes carrying two nozzles as shown. This is another of the advantages of the vertical shaft Pelton wheel, viz. that it enables more than two nozzles to be arranged around the periphery of the wheel, a particularly valuable feature in units of large power where the whole of the energy imparted to the wheel is carried by one or more small jets of water moving at very high velocity. It is probable that the water, after striking the buckets, completes its path over the surface of the buckets and begins to fall into the pit below before the wheel has completed one-quarter of a revolution, and hence for the remainder of the revolution it is running idly without doing any further work. The provision of two or more nozzles reduces this idle period and further gives a more uniform turning movement on the shaft, while since the force exerted by each jet is reduced, being equal to the total force divided by the number of jets, the dimensions

of the buckets can be correspondingly reduced—a valuable consideration in high-powered units. It is hardly necessary to point out that the size of the wheel cannot be reduced, since this depends upon the desired peripheral speed, which must always be about one-half of the speed of the jets, and this depends only upon the head, and not upon the size, of the jet. It is probable, however, that the efficiency of the vertical shaft Pelton wheel suffers from the fact that the water from the top half of the buckets falls back over the wheel on its way to the pit and does not clear them as when the wheel is vertical and

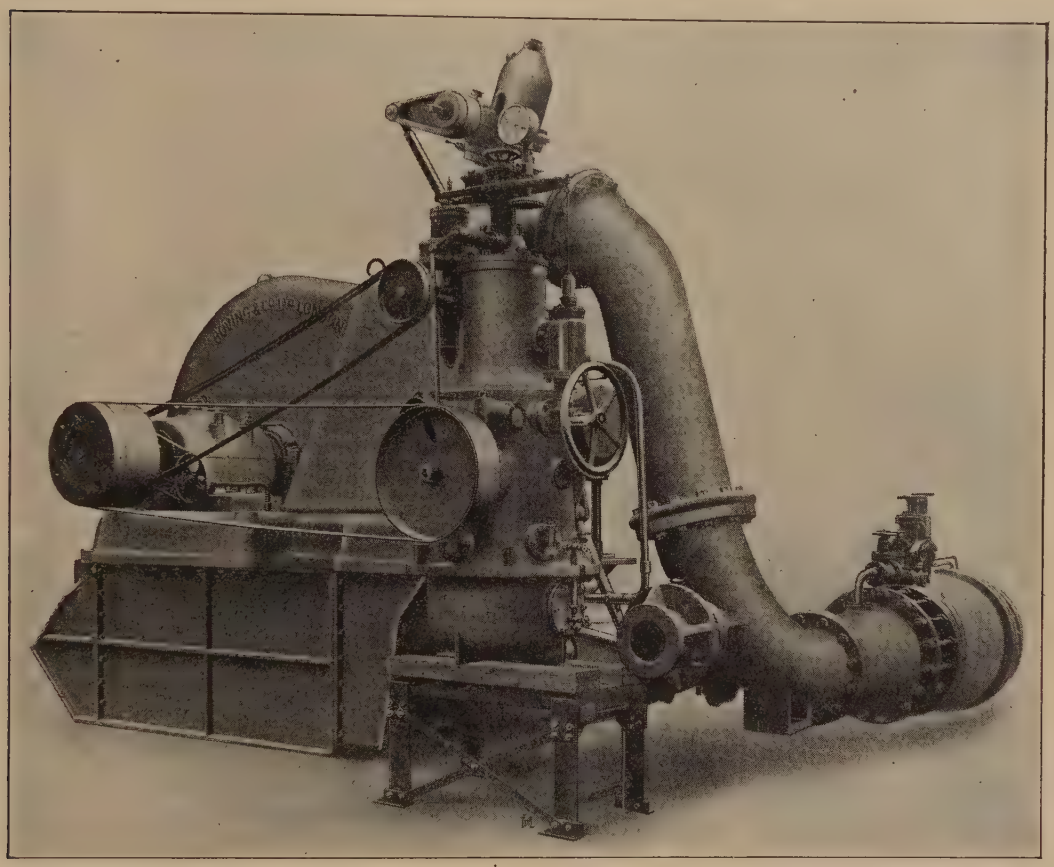

FIG. 203.—Back View of Pelton Wheel of 8,000 h.p., showing Governor Gear and Flange of Coupling. The effective Head is 1,015 ft.

the shaft horizontal; but apart from this, the vertical shaft arrangement for large units has much to recommend it.

The reader must not imagine that in the foregoing brief survey of the subject of water power engineering we have dealt with every phase of the subject. Sufficient has, however, been shown to indicate the methods whereby the never-failing sources of energy which Nature has presented to us in the form of waterfalls and rivers are being harnessed. Particularly should the above description indicate the progress which has been made in this country to utilise such meagre amounts of water power as are available. There is, unfortunately, a tendency to belittle the possibility of utilising water power when only a low head is available. This is to be deprecated, because in a country such as this, where mighty waterfalls and mountainous streams are very few and far between, the low head represents the only possible source of water power that is available, and if we are to make

any use at all of these unfailing gifts of Nature, they must be developed to the fullest extent possible. It is certain that there are very many localities in this country where low-head water power could be developed, although the fact that coal is so easily obtained from our soil makes us slow to look elsewhere for our sources of power.

There is another reason why the low-head water power schemes should be developed, and that is because the knowledge thus acquired will assist us to solve the vast and fascinating problem of harnessing the tides. The chief difficulty in the way of a successful solution of the problem of harnessing the tides is, however, not so much due to the fact that the available head is, generally speaking, very low, as that this head is only available for brief periods during each day. Consider, for example, a place where the average tide is about 20 ft., i.e. the difference of water level between mean high water and mean low water is 20 ft. If it were possible to fill a large reservoir at high tide, then after the tide has fallen, say, 10 ft., it would be possible to pass the water from the reservoir through turbines back to the sea, at such a rate that the level of water in the reservoir fell at the same rate as the tide, thus maintaining a constant head of 10 ft. between the reservoir and the sea. But this could only be done for a period of about three hours, after which the tide would turn. In this way, therefore, we could generate power for three hours out of every twelve—or, to be exact, out of every twelve and a half, since the tide takes on an average about twelve and a half hours for each complete ebb and flow. If, however, we had another large reservoir of equal capacity to the first which was emptied by the tide and kept empty as the tide rose, then after the tide had risen 10 ft., it would be possible to allow the water to pass through the turbines into this second reservoir until the tide was high, i.e. for a further period of about three and a quarter hours, during which power could be developed. Other systems of reservoirs could be mentioned whereby different levels could be created as the tide rose and fell, but it will be apparent that the great difficulty with any tidal power system is to maintain a constant supply of power.

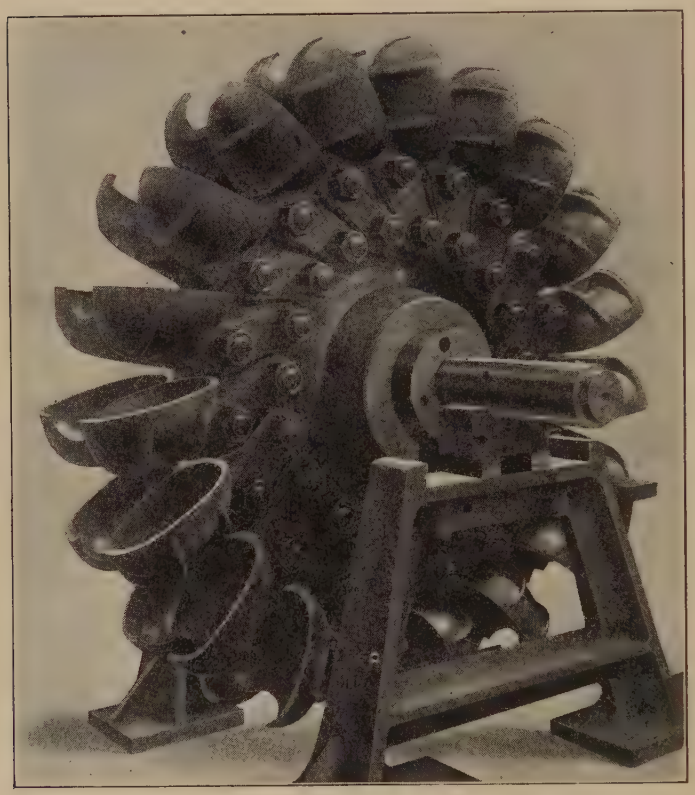

FIG. 204.—Pelton Wheel for Water Turbine of 8,000 h.p., showing Shape and Method of Securing the Buckets. Designed and constructed by Messrs. Boving & Co., Ltd.

Many suggestions have been put forward with a view to overcoming this difficulty, and one of the most promising of these is that during the periods when power is available from the tidal flow, a part of the electrical energy generated should be utilised to drive

huge pumps which should raise water to a reservoir situated at a considerable height above the level of the sea. Then when tidal power is not available, the water from the high level can be allowed to flow back again, driving water turbines and thus generating electrical energy. A certain loss of efficiency would be entailed in this process, but it certainly gets over the difficulty of having certain periods during each tide when direct tidal energy is not available. It will be realised that such a scheme could only be developed in certain localities where a suitable site is available near-by for the high-level reservoir—that is, where there are high hills immediately adjacent to the seashore. Further it is obvious that the capital cost of such a scheme would be very great indeed. Notwithstanding this latter disadvantage, it is the considered opinion of many famous hydro-

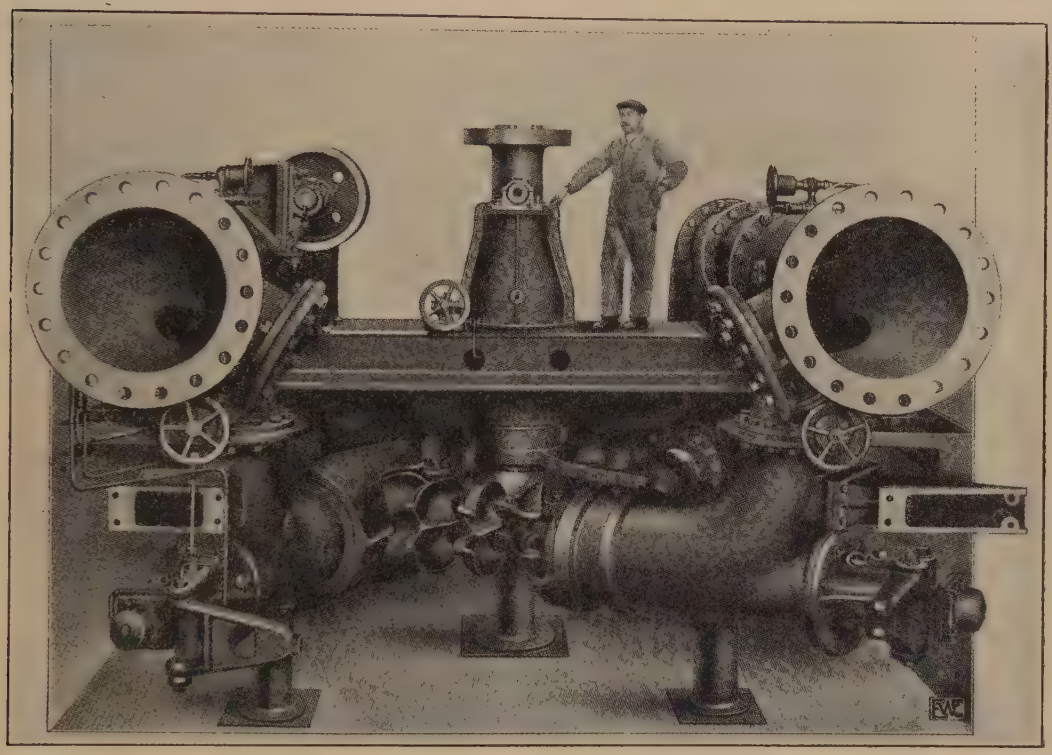

FIG. 205.—Vertical Shaft Pelton Turbine of 15,000 h.p. working under a Head of 836 ft. Designed and constructed by Messrs. Escher Wyss et Cie for the Biaschina Power Station, Bodio, Switzerland.

electric engineers that, provided a suitable site can be found, the scheme is entirely practicable, and that the electrical energy could be generated so cheaply—the fuel costs being *nil*—that a satisfactory return might be obtained on the capital expenditure. It is of interest to note that the famous scheme for harnessing the tides on the River Severn, about which so much was said soon after the conclusion of the Great War, was based upon this idea. This scheme was abandoned owing to the economic depression which soon followed the short period of artificial prosperity engendered by the war; but in view of the possibility of the scheme being again brought forward, we may give a brief outline of the proposals as originally developed in 1920.

The official details of the scheme, as published by the Ministry of Transport, showed that it was considered possible to provide over 500,000 h.p. during a ten-hour day with a peak load capacity of 1,000,000 h.p., at an estimated cost of a little over one half-

penny per Board of Trade unit. The hydro-electric power installation would have been the largest in the world, and it would have effected a saving of from three to four million tons of coal per annum. In addition, the scheme would have created a gigantic locked basin on the Upper Severn over 27 sq. miles in extent, a large portion of which would accommodate ocean-going vessels of the largest size at all states of the tide. One difficulty, however, presented itself in dealing with the River Severn, namely, there are about five times as much water power available during a big spring tide as during a small neap tide, the alternation of spring and neap tides occurring, of course, about once a fortnight. Hence it was necessary to store up energy, not only against the hourly flow, but also against this fortnightly variation in the tide if the maximum amount of power was to be developed. The proposed scheme was substantially of the form indicated above, except that the high-level reservoir had to be made of great capacity in order to meet the extra demand during periods of neap tides. It was proposed, therefore, to place a barrage, i.e. a large dam, across the River Severn with sluices and low-head water turbines for utilising the tidal flow, and an artificial salt water lake high up in the valley of the Wye, with a pumping station and high-head turbine power house on the banks of the tidal portion of that river.

The site chosen for the Severn barrage was close to the line of the Severn Tunnel, where the estuary is about two and a half miles wide, the greater part of the river bed being exposed at low tide, with low-lying country on either side. On both sides are great stretches of rock, while between them is a channel rather wider and much deeper than the River Thames at Westminster, which would form a natural tail race for carrying away the water from the turbines installed in the walls of the dam. At high tide, the water which has flowed past the dam into the estuary of the river would be trapped by the sluices, and when a sufficient difference of level is reached between the river and the impounded water, it would be allowed to flow through the turbines. It was estimated that, in view of the enormous volume of water trapped in the natural reservoir formed by the dam it would be possible to generate upwards of 1,000,000 h.p. each tide, the amount being greater, of course, during the spring tides than during the neap tides. A part of this power would be used to pump water from the low level of the River Wye through a tunnel 40 ft. in diameter up to a large reservoir in the Wye Valley formed by a dam across the lower end of the valley not far from the ruins of Tintern Abbey, electric power generated by the low-level turbines at the Severn barrage being transmitted through cables for operating the pumps in the Wye power house. When, however, the demand for current is greater than the Severn turbines can supply, or when these are idle on account of the state of the tide, pumping in the Wye power house will cease and the water will flow back through the tunnel, driving high-head water turbines which are coupled to electric generators, thus supplying electricity to the system.

In order to develop as large an amount of power as possible from the tidal flow, it was decided that the low-head turbines should be put into operation soon after the tide began to ebb, and that they should continue working through the hours of low water until about half flood tide, which, it was estimated, would give a working period of about seven hours followed by five or six hours during which no power would be obtainable from the tide. The head of water available at the turbines would during the seven hours' working period vary from 5 ft. to a maximum of about 30 ft. at periods of spring tide back again to 5 ft. This would mean that the speed of the turbines would vary, and it was determined that, using Francis wheels 10 ft. diameter with vertical shafts, the limits of speed of the turbines would be between 40 and 80 revs. per min. To enable a constant voltage to be obtained from the generators, these were to be direct current machines driven through $7\frac{1}{2}$ to 1 step-up gearing giving a speed at the generators of 300 to 500 revs. per min. and separately excited, the voltage control being obtained by regulating the

excitation. The generators were to have an average output of 1,300 kw. at a constant pressure of 525 volts, and the direct current was subsequently to be converted to alternating current at 330 volts by means of large rotary converters, the voltage being subsequently raised by static transformers to 60,000 volts. At this pressure it was proposed to transmit the energy to the industrial districts of Wales and the Midlands, while some would be used in the immediate neighbourhood, where it was hoped that industries would arise as they have done at Niagara. At the Wye pumping station the pumps were to be driven by alternating current motors, the pressure of the supply being 2,200 volts, while in order to save the cost of installing two sets of machinery, it was proposed to combine in one unit a motor for driving the pump and a generator to be driven by a turbine, the latter being in operation when power is being obtained from the Wye Valley reservoir.

The great stumbling block to the realisation of the scheme was, as already indicated, the capital cost. While no figures were given by the Ministry of Transport, it was estimated that to complete all the works involved and to equip the two power stations would involve a capital expenditure of £40,000,000. It will be recognised that to pay interest on this enormous sum, and at the same time to allow for depreciation, etc., would demand a huge annual revenue. The high rates of interest which ruled after the war led to the scheme being abandoned, despite the fact that it was proposed to obtain the support of the Great Western Railway and of the local authorities by the provision of a railway and roads across the barrage. It is, however, not improbable that the scheme may again be brought forward when economic conditions permit, and no doubt it will then be possible to reduce the capital cost of the scheme as well as the interest on the loans. Meanwhile our advancing knowledge of the science of utilising water power should tend to give us improved methods whereby the tidal power which is undoubtedly available at the mouth of the Severn could be utilised. In any case, the working out of the details of the scheme which we have briefly set out above served the great purpose of showing that it was possible to obtain a continuous supply of electrical energy from the fluctuating power of the tides. Since that time many other schemes have been worked out, all of which bear the stamp of practicability, at least from an engineering point of view. The time may come when economic conditions may render it advisable to put these and other schemes into operation. That power can be generated from the tides is undoubted, but time alone can tell whether this, together with water power obtained from the inland rivers, will form the main source of power of the future.

END OF VOLUME III

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